TRANSACTIONS

AMERICAN SOCIETY

HEATING AND VENTILATING

ENGINEERS

VOL. 27

1921

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CHAMPLAIN L. RILEY
President 1921
of
AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING Air Conditioning ENGINEERS

VOLUME 27

TWENTY-SEVENTH ANNUAL MEETING
PHILADELPHIA, JANUARY 25-28, 1921

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OF

American Society of Heating and Ventilating Engineers

No. 580

THE TWENTY-SEVENTH ANNUAL MEETING

1921

THE Twenty-seventh Annual Meeting is one that established a new precedent in the history of the American Society of Heating and Ventilating Engineers, namely, that of holding the professional sessions of the Meeting outside of New York. For 26 years the Society has held its Annual Meeting continuously each year in New York City, until the feeling had developed that an annual meeting might not be successful if held elsewhere. When as a result of the cordial invitation of Eastern Pennsylvania Chapter, the decision was made to deviate from former precedent, and the professional sessions of the Annual Meeting were held in Philadelphia, Pa., with the result of an attendance, interest and enthusiasm more than double that in evidence at any previous Meeting, the conclusion has very generally been arrived at by mutual consent amongst the membership that the departure was justified and that the *experiment* may be worthy of repetition.

The net result of the Meeting in Philadelphia was to indicate that the past year of activity in the Society has been fruitful of great progress and remarkable development. The range of technical subjects discussed, as well as the interest taken in the work of the many active Committees of the Society, showed decidedly that the membership is alive to the needs of the profession and determined to live up to the highest sense of its duty in this respect. The remarkable total attendance of 532 members and guests registered, which more than doubles any previous record, is an eloquent testimonial as to the character of the Meeting. Of this number, 261 were members of the Society and 271 guests; of the latter number, 96 were ladies.

The opening session of the Meeting, which was the Annual Business Meeting, was held in accordance with the requirement of the Constitution in New York City, at the United Engineering Societies Building, 29 West 39th Street. At the close of the business session, the Meeting adjourned, in accordance with the arrangements made by the Council, to Philadelphia, where the remaining five professional sessions were held amid surroundings that were rendered unusually pleasant by the delightful host—Eastern Pennsylvania Chapter.

PROGRAM

FIRST SESSION

Tuesday, January 25, 10 a. m.

Engineering Societies Building, New York, N. Y.

Business Session:

Announcement of Quorum.
Appointment of Tellers of Annual Election.
Report of President.
Report of the Council.
Report of the Secretary.
Report of the Treasurer.
Reports of Committees:
 a. Executive Committee.
 b. Finance Committee.
 c. Publication Committee.
 d. Membership Committee.
 e. Committee on Revision of Constitution.
Unfinished Business.
Report of Tellers of Annual Election.
New Business.
Adjournment of Meeting to Philadelphia for Professional Sessions.

SECOND SESSION

Wednesday, January 26, 10 a. m.

Clover Room—Bellevue-Stratford, Philadelphia, Pa.

Professional Session:

Addresses of Welcome.
President's Address.
Resume of Business Session at New York.
Reports of Technical Committees:
 a. Committee on Code for Testing Low Pressure Heating Boilers.
 b. Committee on Standard Code for Testing Heating Systems.
 c. Committee on Steam and Return Main Sizes.
 d. Committee on Schoolhouse Standards.
 e. Committee on Standard Method for Testing Air Washers.
 f. Committee on Furnace Heating.
Discussion of Code of Ethics.

THIRD SESSION

Wednesday, January 26, 8 p. m.

Clover Room—Bellevue-Stratford, Philadelphia, Pa.

Chimney Session:

Discussion of Model Chimney Ordinance proposed by the National Board of Fire Underwriters.

Paper:
 Radial Brick Chimneys, by W. F. Leggo.

Paper:
Steel Smoke Stacks, by W. E. Goldsworthy.

Paper:
Some Comparative Tests of 16 in. Roof Ventilators, by Dryden, Stutz and Heald.

Fourth Session

Thursday, January 27, 10 a. m.

Clover Room—Bellevue-Stratford, Philadelphia, Pa.

Research Session:

Report of Chairman of Committee on Research.

Paper:
Theory of Dust Action, by O. W. Armspach.

Paper:
Efficiency of the Palmer Apparatus for Determining Dust in Air, by S. H. Katz, E. S. Longfellow, A. C. Fieldner.

Paper:
Physiological Heat Regulation and the Problem of Humidity, by E. P. Lyon.

Paper:
A Study of the Infiltration of Air in Buildings, by O. W. Armspach.

Paper:
The Transmission of Heat Through Single-Frame Double Windows, by A. Norman Shaw.

Fifth Session

Thursday, January 27, 2 p. m.

Clover Room—Bellevue-Stratford, Philadelphia, Pa.

Fuel Session:

Paper:
Pulverized Coal for Power Plants, by F. A. Scheffler.

Paper:
Briquetted Coal for Household Fuel, by J. H. Kennedy.

Paper:
Economizers, by W. F. Wurster.

Topical Discussion:
Pulverized Peat Fuel.

Sixth Session

Friday, January 28, 2 p. m.

Clover Room—Bellevue-Stratford, Philadelphia, Pa.

Professional Session:

Paper:
Forced Hot-Water Circulation Heating System, Girard Estate, Philadelphia, Pa., by Robert Hughes.

Paper:
Accelerated Hot-Water Systems, by A. J. Wells.

Paper:
Influence Upon Boiler Economy of Continuous Firing with Highly Volatile Fuel Without Intervening Cleaning, by A. B. Reck.

Paper:
Design of Large Boiler Plants, by J. Grady Rollow.

Paper:
Proper Selection of Hot-Water Heating Devices for Domestic Service, by A. Buenger.

Paper:
Method of Utilizing Heating Systems for Cooling Rooms in Summer, by A. M. Feldman.

CODE OF ETHICS

An explanation of the value of an engineers' code of ethics was given by A. G. Christie, who acted as Chairman of a Committee of the American Society of Mechanical Engineers working with the representations of other leading technical societies to draft a suitable code. He pointed out its significance especially in relations of engineers to the public and said that the final code should be one to which all engineers can subscribe. After a full discussion of the subject, the members of the Society voted to appoint a Committee to act with the other societies and report at the next meeting.

A. G. CHRISTIE[1]: I wish in opening my remarks to express my great appreciation for the privilege of talking to you on the code of ethics. A code of ethical conduct it seems to me is a very serious matter for engineers, because it affects their professional standing, particularly with regard to the public.

What is a code of ethics? Our idea of it is that it is a statement of right, honorable and just conduct on the part of the engineer in his relation to the public, to his fellow engineers and to his employers, employees or clients, as the case may be.

There are questions coming up continuously in which some expression of opinion on ethical conduct is desired. Since our committee in *The American Society of Mechanical Engineers* has been formed and has been acting we have had informally presented to us a great number of cases asking us to express our opinion on ethical conduct on the part of engineers involved. In almost every case there was absolutely no intention apparent of wrong doing, but the engineer was in a dilemma and did not know just exactly what was the right thing to do.

WHAT SHOULD BE DONE?

Here is one case about which I can speak. A consulting engineer and his firm had been engaged to build a large establishment and he incidentally owned certain patented machines which might be used in some new equipment that was to go into this new establishment. The question arose between the owners of that establishment as to whether he should not repay to them his royalty on those patents which were being built and furnished by a separate company from his consulting engineering company. What is the correct ethical conduct under those conditions?

Here was another case: A young engineer had a number of patents on appliances for machine tools. He was offered employment by a concern at what we considered a rather moderate salary. He accepted the employment. There was nothing further said until he went to take up his work, when he was asked to sign a contract by which he signed over all patents which had been taken out previous to his employment, to the company which was employing him. That did not seem right. It seemed to be quite apparent that there was a breach of ethical conduct on the part of the people who were employing him.

Our idea is that we need to prepare a code of ethics stating some of the

[1] Associate Professor of Mechanical Engineering, Johns-Hopkins University.

things which an engineer should and should not do and then also to provide a committee on professional conduct which would act as a kind of consulting engineer's board.

HOW DOES PUBLIC REGARD ENGINEERS?

We also want to look at the question of a code of ethics from another point of view. Is engineering regarded as a profession by the public? Are engineers put on all sorts of city boards and state boards that are to pass on highways, new tunnel constructions and such? No; as a rule they will put on lawyers, doctors, business men, contractors and others. When the technical side has to be worked out they call on the engineer, who is generally regarded as a technician, but they do not look for the same professional service spirit in the engineer that they do in these other professional men. That is not right. We recognize among ourselves that we are fully as capable of serving the public as these other men, but the public does not recognize this service spirit at the present time among engineers.

Regarding other professions, Mr. Rice made the statement in New York recently that physicians had a code of ethics as far back as 500 B. C., and they practically subscribe to that same code today when they take up their professional duties. Since the physicians and the lawyers have a recognized code of standards, they are recognized by the public and given this public work and public service.

CODE RAISES PROFESSIONAL STANDING

I maintain that if the engineers can prepare this code of ethics and conscientiously subscribe to it, we will get the same public recognition and the same attitude towards service that the others have. We will also get a really professional standing in the community rather than simply a technical standing, as we have at the present time.

Our committee in *The American Society of Mechanical Engineers* of which I happen to be chairman, was formed very informally, and men were picked out who practically knew nothing much about a code of ethics. After considerable discussion and advice, we decided that the first thing to be done was to prepare a code for the whole engineering profession;—a code for the Mechanical Engineers alone would amount to practically nothing, as it would not have any standing with the public. Our first recommendation to the Council was that they would take the matter up with the Engineering Council at Washington. The Engineering Council, when the matter was presented to them, were just about at the point of dissolving, so they referred it back to us as they felt it could be better handled through the individual societies. We therefore approached the other societies to see how they felt in the matter and we have not gotten through our work along that line. *The American Society of Civil Engineers* has appointed two members to cooperate with us in forming a general code. The *American Institute of Electrical Engineers* has its committee on professional conduct that is also working with us, and one or two of the other societies have the matter under consideration. In this Society we have had very helpful suggestions from one of your members, Thornton Lewis.

At the last meeting of the Mechanical Engineers in December we submitted to them a report in which we offered a tentative code. There was a great deal of comment and very helpful criticism, which has enabled us to see a number of things which we can easily correct.

COMMITTEE ON PROFESSIONAL CONDUCT RECOMMENDED

One of the things that we did find was that we seem to have provided the proper way of administering the code. I will just comment very briefly on our method. It is suggested that the administration of the code of ethics in the societies—this would refer now to other societies than the Mechanical Engineers—ought to be put up to a standing committee on professional conduct; the duties of this committee to be to interpret the code, to interpret any cases of questionable ethical conduct on the part of members that may be submitted to them, and to refer those interpretations to the Council or other governing body of the society; then the Council may approve these interpretations or take such other action as may seem necessary or just, and those interpretations to be published in the publications of the society taking up the cases. We suggested the method of appointing this committee and we also suggested that the By-Laws be modified in such a way that this committee can call in subcommittees to advise them.

The Council shall have the power to act on any recommendations of the committee on professional conduct; either, first, to censure by letter the conduct of the member who has acted contrary to the code or for breaches of that character; or, second, to cause the member's name to be stricken from the roll of the society.

I shall not take up your time to read the code itself. We had about fourteen clauses but were asked to break it up into fifteen in order to improve upon the grammar and the phrasing.

Another objection was that it was too long. One man said that he would prefer to have about six clauses in the code of ethics that he could have printed, framed and hung up in his office. That is very desirable and is one of the things we wish to effect if we can possibly do so.

My personal feeling is that I want a few good strong statements on what an engineer will be expected to do in his relations to the public, to his fellow engineers and to his employers or clients; and we can express those special clauses as a kind of suggestion at the end of our report to the committee on professional conduct that will guide them and will guide others who will take up the matter and who may have to write a code at other times. But we would like to have a short code if possible to submit to all societies.

This final code should be one which all engineers can subscribe to and which will place engineering on a professional basis. It is going to take a good deal of consideration and thought, to get everything straight in that Code, and the request that I have come here before you to make, is that you will help us in preparing it. I should like to see some of your members appointed to act on the Committee to consider this joint Code.

DISCUSSION

S. A. Jellett: I think it is of very great importance that all engineering societies join in this code of ethics. The trouble has been all along, not that men did not intend to do right, but that they are guided very largely by their clients in many cases. They are paid by one man and have to arbitrate between the man who pays them and the man who does not pay them and who sometimes is trying to get the better of him by short circuiting on his contract.

I frequently act as arbitrator on disputed contracts. In one particular instance the chairman of a scientific institution got into trouble with his contractor. He was an exceedingly arbitrary man. I was selected by the builder and by the electrical contractors to represent them in the arbitration. It was a very complex system of electrical work that had been partly installed when orders came to stop and substitute a new patented scheme which had been submitted to the president of the institution but had not been submitted to the architect or his consulting engineer. When the order to stop work was issued some $1,200 or $1,400 worth of work had already been installed and the contractor was asked to name the additional cost of substituting this new system. It amounted to something like $2,800. He was told his bid was rejected and was ordered to proceed with the work under the arbitration clause of the contract; the amount to be paid him would be determined after the work was completed. He did that. After the work was completed the matter was put to arbitration. I was selected by the general contractor and the electrical contractor; the institution selected its own consulting engineer as the second one and we two in turn selected the third man, who was a well-known engineer and had no connection with it. At our first meeting the engineer representing the institution made his statement, which was, "We are here to determine whether we shall pay these people the amount of their bid for additional work or how much less we shall pay them." I said, "That has nothing to do with it. There is no contract. It is gone. We are here to determine the value of the work this man has done. The stopping of his work originally under the contract ended the original agreement. We are now to determine the value of the work." The other man couldn't see that at all. I said "My people bid but their bid was rejected. Therefore no contract exists and the arbitration is on the value of the work." The referee agreed with me that that was right. The other arbitrator was unwilling to go ahead. He left us and consulted his chief (also consulted an attorney, he told me afterwards), as to the legality of my position. We came together shortly afterwards and he admitted that I was right, that there was no contract, that the value of the work must determine it.

This engineer had no training in business matters at all, otherwise he would not have taken the position he did. And he was practically under the domination of his chief; that is, he knew it would be displeasing to his chief to take any such position. The chief's idea was that he was going to save half the cost of that extra work. I then produced a sworn statement of the actual cost of the work, list and schedule of materials and so on. I said, "Now let's agree what is a fair percentage for overhead expense." Then a percentage for profit. These were added to statement of costs. Instead of getting $2,800 the contractor got over $5,300 and he was entitled to it.

If that engineer had had a business training he would have recognized at the very start that when a man abrogates a contract and tells you to go ahead, "We will arbitrate its value," and the work goes on, the fair value is the only consideration. If a code of ethics was in use then the owner's attention could be called to it and I believe it would be a good thing.

WILLIAM H. MCKIEVER: As a member of both Societies I am very much interested in it, and I am particularly interested as being one of the early ones in *The American Society of Mechanical Engineers* that suggested the necessity of a code of ethics.

The Chairman of the Committee in *The American Society of Mechanical Engineers* has stated that they wanted something to put the teeth in the code of ethics. There is nothing that can do this except the enactment of a national law that will give an engineer standing. The reason that the engineer has not such standing and the reason that the lawyer has, is because he has the protection of the law. We have no such protection.

I went into a drafting room one day and found a man wearing a badge of *The American Society of Mechanical Engineers* that I knew was not a member of the Society. If I went and wrote out a prescription and put my name to it the law would take care of me, but that fellow was sailing under a false flag and could not be touched. No code of ethics will ever have any teeth in it unless it has behind it the protection of the law.

We must start right at the college. When a man graduates in any division of engineering he must subscribe to a code of ethics the same as a man does when he takes his oath in medicine and law. Unless that is done it means absolutely nothing.

I was very glad to hear Mr. Jellett's story and would say that the training of engineers in college should be broadened out more, so that the engineer not only becomes an engineer but something of a lawyer besides, with a broad reasoning of what is in this code, the innermost side of the question involved.

THORNTON LEWIS: As to a code of ethics and the introduction of such a code to the man as he leaves college (at which time he does not know which branch of the engineering profession he is going into) it seems to me necessary for the engineering societies to get together and formulate a common code of ethics so that it will be alike for all engineers. I would like to move you that it is the sense of this meeting that the President of our Society appoint a Committee to cooperate with the Committee of *The American Society of Mechanical Engineers* and sister societies in formulating a common code of ethics for all engineers.

The motion was seconded and carried.

FRANK K. CHEW: We do not often have so many members from all over the country meeting together as we do here; but in the former years, when we had a comparatively small number, there was a fine chance to see each other and talk about ethics and to see that the men who were engaged every day in making money from the sale of apparatus were separated a little bit from the men who were making money from the designing of apparatus. Ethically there is a very strong difference. The self-consciousness is supposed to be more pronounced when you have got something to sell of a material nature than when you are held responsible for what you recommend and for which you accept a fee.

At this meeting some of the old-timers thought that the dignity of the Society had not been carried in the debate to the extent that it should be, and that the accent has been put on the commercial side more strongly than is good.

An engineering society is different from any other association in which we engage. In the talk here some people alluded to this association. They failed to differentiate between an association and a society. Now that may be splitting hairs, in some people's minds, but I want to impress this body of men here that when we had 150 or 200 members we were criticized severely, mud was thrown at us in plenty and only a little of it stuck. There will be more mud thrown at us now because we have now 1500, or will have soon. That 1500 has a responsibility and I want to point it out. It is a good deal like saying everybody should have a living, you are out for yours. You want to eradicate some of the things which are really strongest in your line. Otherwise the Society cannot come up to that plane which the Research Bureau has brought up this morning before you.

It is very clear to me that the world is looking toward us, and when we have the opportunity we must not be dragged down to the muddy level because of our close commercial connections; we must live up to a higher plane. And any member of the Society who is going to do something different, he must stop and think a minute and think whether he is doing his level best or whether he is pursuing his every-day course.

No. 581

DISCUSSION OF CODE FOR TESTING LOW PRESSURE HEATING BOILERS

The Code for testing low-pressure heating boilers, as originated in 1918 and revised at the Annual Meeting of the Society in 1919 (see Volume 25 of the Transactions, 1919, page 143), has been given wide usage during the past two years. One of the first organizations to adopt the revised Code of 1919 for its official use in the testing of heating boilers was the U. S. Bureau of Mines at its Experiment Station at Pittsburgh., Pa., under the direction of John Blizard, fuel engineer. The Code has there been thoroughly tried out in its application to the testing of all classes of low-pressure heating boilers and also for their use on all classes of fuels. The result of the experience gained in connection with the Bureau of Mines tests has led Mr. Blizard to prepare the following statement offering constructive suggestions for further revision of the Code.

Bureau of Mines, Experiment Station, Pittsburgh, Pa.

October 22, 1920.

To the Committee on Code for Testing Low Pressure Heating Boilers:

In a general way I find the Code too brief, nor is there to be found anywhere precisely by whom the Code is intended to be used. I presume that it is intended to be used for commercial boiler tests. If it is intended to be used for commercial boiler tests, and it is necessary to put in so many details as a thoughtful committee has seen fit to include, then it seems to me that it would have been wise to have gone a step farther and written a general preamble on the principles involved in testing boilers. If this preamble were well written, the Code would appeal to a wider field.

The present Code appears to me to be too condensed to be useful by those not familiar with the testing of boilers, and to contain little or nothing which will aid those familiar with the testing of boilers.

Whatever changes you may see fit to make in your Code, I hope you will endeavor to include those changes which exclude some antiquated and useless items, such as an "evaporation from and at 212 deg." I know you are anxious if possible to keep but one unit of energy in the report, namely, the British thermal unit. For the same reason I should like to see the unit now used for the rate of supplying energy either in horse power or radiation disappear, and to see in its place, British thermal units per hour.

Yours very truly,

(Signed) John Blizard, *Fuel Engineer.*

CRITICISM OF CODE FOR TESTING LOW PRESSURE HEATING BOILERS

(Code published in the TRANSACTIONS, Vol. 25, 1919, pages 143-147)

Apparatus and Instruments (p. 143): No mention of feed tank from which the feed pump draws; this tank should be provided with a fixed-point gauge or float, and water brought to some level in the tank after definite intervals of time.

Note should be made that the weighing or measuring (by volume) tanks should hold water sufficient for ten minutes supply to the boiler. When measuring by volume, the upper level of the water should have a small sectional area to reduce the error when refilling either to a fixed mark or to overflowing. The measuring tanks should have conical bottoms to ensure emptying all the water.

The volume-measuring tanks should be cylindrical to avoid changes in volume and be calibrated by filling with known weights of water at the temperature of the feed water.

Small Accurate Scales: Why are these scales to be *accurate,* while the feed-water scales are merely to be *tested*?

Draft Measurements (p. 143): Change to read: Draft should be measured preferably by an inclined U-tube, which shall be accurately set up and inclined with both legs in the inclined plane, so that the draft can be read to one-hundredth of an inch of water, by measuring the distance between the two menisci of the fluid in the tube. The density of the fluid must be allowed for in inclining the tube, so that the draft is obtained by multiplying the scale reading by a whole number, preferably a multiple of ten.

Moisture in Steam (p. 143): How is the efficiency of the steam separator to be tested?

Gas Analysis (p. 144): Add a statement that care must be taken to see that no air leaks into the sampling pipe; also that a continuous sample of the gas must be drawn off at a constant rate to insure getting a mean average of the composition for the trial.

Weather Reports (p. 144): Very doubtful if of any value except to partially account for the draft. The loss due to heating the steam entering with the air and leaving with the flue gas is exceedingly small and the change in heat content of the steam for the same boiler gauge pressure, but different barometric pressure will not amount to more than about three heat units in 10,000 heat units.

RULES FOR CONDUCTING EVAPORATIVE TEST

The Primary Object of Test (p. 144): Very few—possibly no boiler trials have been conducted to determine *primarily* the evaporation per pound of *dry* fuel consumed. Usually the primary object is to find the evaporation per pound of fuel fired. But the evaporation per pound of fuel need not of necessity be the primary object—it may easily happen that the primary object is to determine whether the boiler as installed will generate steam at the maximum rate required, and the ratio of the steam generated to fuel fired may be a secondary consideration.

Duration of Test (bottom of p. 144): The Code here appears to sanction a test of 12 hours duration, though qualifying it by suggesting that longer tests be run. Obviously the test should last until some minimum quantity of fuel has been burned per square foot of grate. If, for example, this quantity be fixed at 120 lb. per sq. ft. per test, then if 3 lb. of coal per sq. ft. per hour be burnt, the trial will last 40 hours, and if 8 lb. per sq. ft. per hour be burnt, the trial will last 15 hours. Suppose the continuous-fire method of starting and stopping be used, that a fuel bed 10 in. deep be carried, that due to error of judgment the mean fuel bed at the end of the test be 1 in. shallower than at the start of the trial, and that at the start 1 cu. ft. of fuel bed contain the equivalent of 24 lb. of coal while at the end it contain the equivalent of 20 lb. of coal; the error for the trial in equivalent pounds of coal per square foot of grate would then be:

$$24\ (1/12) + 8/12\ (24 - 20) = 2 + 2\ 2/3 = 4\ 2/3 \text{ lb.}$$

or nearly 4 per cent more coal will have been burned than will have been recorded by weighing.

Nor is error eliminated by using the new-fire method, since there must be considerable sensible heat energy remaining in the incandescent fuel dumped at the end of the trial, which is not accounted for, and the chemical energy content of the fire bed dumped at the end of the test cannot be determined accurately unless sampled and the sample burnt in a calorimeter.

New Fire Method (p. 145): The Code permits the use of the new fire method of starting and stopping tests on small boilers using anthracite. A short trial carried out when using this method will show what happens when the furnace is *started* for the season, but may mislead people into imagining that the results of this test will represent the normal operation of the furnace when the fire pot contains a larger proportion of ash and partially burnt fuel. The difference between the composition of a new and old fire bed materially affects the necessary draft to burn the fuel and the general operation of the furnace. It is doubtful if the new-fire method should be approved of.

CONTINUOUS FIRING METHOD

Starting and Stopping (p. 145): It is recommended in the Code that the boiler be operated for at least one hour before starting test. This should read somewhat as follows: "The boiler shall be operated before the test until it is necessary to clean the fires. After cleaning the fires, the boiler shall generate steam at the normal test rate for one-half hour or more. During this half-hour the fuel fired, water level, steam pressure and temperatures of flue gas shall be observed and recorded. At the end of the stipulated time (one-half hour or more), the thickness of the fuel bed shall be estimated or measured," etc., as in the Code. Before stopping the test, the boiler operation preliminary to starting should be duplicated as far as possible, after the final cleaning of the fires.

The objection to starting the test after only one hour's preliminary steaming lies in the possibility, within the laws of the Code as now written, in starting with a new fire an hour before the test. When a fire is cleaned when it *needs* cleaning, the residue on the grates after cleaning contains more ash and partially consumed fuel than does the residue after *cleaning* a new fire which requires no cleaning. Thus for some hours before starting, the boiler would not be operating normally and the residue after *cleaning* a clean fire at the start, would be richer in combustible than the residue after cleaning a dirty fire at the end of the test, and the apparent thermal efficiency of the boiler would be higher than the actual thermal efficiency.

GENERAL

Samples of Coal and Residue from Ash-Pit: It is stated that the samples of coal shall be preserved for subsequent determinations of *chemical composition*, and that the residue from the ash-pit *shall be reduced by quartering*—for subsequent *chemical analysis; chemical composition* and *chemical analysis* should be defined. The disposal of the refuse removed from above the grate is not referred to.
Report Form (p. 146):

1. Rating. Is there a standard method of rating?
9. Add: B.t.u. per lb. as fired.
10. Define *Fuel Capacity*.
11. Define *Fuel Available;* give units in which measured.
12. State how this is to be determined.
15. Unnecessary item.
16. How is this determined?—far better leave out items 16 and 17; they mean little if anything. Presume *fuel* recovered from refuse is anything that looks black, but all that is black has not the same calorific value; it may be greater or less than the calorific value of the fuel fired.

20-21. Define ashes and clinker.

22. This item might mean many things—would leave it out.

25-26 and 27. Give units—deg. fahr.

28-29 and 30. Give units—inches of water.

31-32 and 33. Cannot be determined from the draft measurements alone, as the mean density and velocity of the gases must also be known; better refer to these items as "The difference in draft between ash-pit and furnace," etc.

34-35. Give units—lb. per sq. in.

40. Leave out this item.
41. Substitute "heat transferred to water per hour" in 1000 B.t.u.
42. Substitute "heat transferred to water per hour per lb. of coal fired" in 1000 B.t.u.
43. Leave out this item, far better to account for heat in heat balance alone.
44. This is the same as item 41 as amended above, according to the definition, though "capacity, total available B.t.u." is a vague title.
45. Apparently the same as item 44, but is better defined.
46. Substitute "heat transferred to water per hour per sq. ft. radiation rating of boiler." How is the *radiation* rating measured?
47. Leave out this item; let all losses determined be represented in a balance sheet. It should be noted that neither the efficiency of the

grate nor of the boiler and furnace, nor a proper heat balance can be determined without determining the composition of the fuel, refuse and *flue gas*. No items are included in the report form to show the composition of the fuel, refuse and *flue gas*. If from these tests a heat balance is to be made out then the above items should be included in the report form proper, and not on the auxiliary sheet only.

A suggested form for test of low-pressure heating boiler is given below.

EXTRA SHEET—INSERT FOLDER

This sheet contains some particulars which should be reproduced on the main report sheet, as flue gas composition, *residual* composition. What is *residual*?

If the composition of the residual is determined why not determine the composition of the ash and refuse? And why if unusual care is taken to determine the sulphur in the residual is not the ultimate analysis of the coal determined?

It would be well to add columns in which to record the ash removed from the ash-pit, and from above the grate.

To be consistent with the rest of the Code and to be accurate, lb. should be constituted for lbs.

Respectfully submitted,
(Signed) JOHN BLIZARD, Fuel Engineer.

REPORT FORM FOR TEST OF LOW-PRESSURE HEATING BOILER

Test. No. ....

Type of boiler: ...., Made by: ....
Kind of fuel: ....
Test made at an output of.................lb. of steam per hr.
Site of test: ....
....

Reference Number	General particulars of trial
1	Duration, from .... to .... hrs.
2	Method of starting and stopping test.
3	Method of firing.
4	Longest interval between firing .... min.
5	Average interval between firing .... min.
6	Diameter of orifice in orifice box.................................. in.
	Particulars of boiler, furnace and grate
7	Kind of furnace.
8	Grate surface (width........ length ....) sq. ft. or diameter
9	Approximate width of air openings in fire bars.................... in.
10	Ratio of air openings to total grate area.................... per cent.
11	Kind of grate.
12	Total heating surface (furnace boundary above bars ....) sq. ft. remainder .... sq. ft. .... sq. ft.
13	Height from grate bars to roof of furnace.......................... in.
14	Volume of furnace between grate and roof of furnace.......... cu. ft.
	FUEL
15	Size ....

16 Ultimate analysis, dry fuel: ....
a. Hydrogen .... per cent.
b. Carbon .... per cent.
c. Nitrogen .... per cent.
d. Oxygen .... per cent.
e. Sulphur .... per cent.
f. Ash .... per cent.

17 Approximate analysis of fuel as fired:
a. Moisture .... per cent.
b. Volatile matter .... per cent.
c. Fixed carbon .... per cent.
d. Ash .... per cent.
e. Fuel ratio, F.C./V.M. ....

18 Calorific value of fuel per lb.
a. As fired .... B.t.u.
b. Dry .... B.t.u.
c. Moisture and ash-free .... B.t.u.

19 Total fuel fired during test .... lb.

ASH AND REFUSE

20 Ash and refuse removed from above the grate during test.......... lb.
21 Total ash, clinker and refuse removed during test from ash pit and grate .... lb.
22 Total ash in fuel fired during test.......... lb.
23 Carbonaceous matter in dry ash, clinker and refuse removed .. per cent.

DRAFT

24 Draft in ash pit .... in. water
25 Draft in furnace .... in. water
26 Draft in stack .... in. water

AIR

27 Temperature of air entering ash pit.......... deg. fahr.
28 Excess air in flue gas, per cent of air used for combustion..........

FLUE GASES

29 Analysis of dry flue gases by volume
a. Carbon dioxide .... per cent.
b. Oxygen .... per cent.
c. Carbon monoxide .... per cent.
d. Nitrogen (by difference) .... per cent.

30 Temperature of flue gases leaving boiler.......... deg. fahr.
31 Lb. of dry flue gas per lb. carbon in flue gas..........

STEAM AND FEED WATER

32 Gauge pressure, boiler .... lb. per sq. in.
33 Gauge pressure in orifice box.......... lb. per sq. in.
34 Temperature of feed water .... deg. fahr.

RATES

35 Heat transferred to water per hr..........1000 B.t.u.
36 Heat transferred to water per sq. ft. of heating surface per hr.... B.t.u.
37 Fuel fired per sq. ft. of grate surface per hr.......... lb.

	HEAT BALANCE PER LB. FUEL AS FIRED	B.t.u.	Per cent
38	Heat transferred to water (and thermal efficiency)...	......	......
39	Heat carried away by steam in flue gases..........	......	......
40	Heat carried away by the dry flue gases..........	......	......
41	Heat lost by not burning carbon monoxide..........	......	......
42	Heat lost by not burning carbonaceous matter removed from ash pit and grate..........	......	......
43	Undetermined heat losses	......	......
	Total (and calorific value of fuel)..........	......	......

NOTES ON CALCULATIONS—LOW PRESSURE BOILER TESTS

44 Excess air present in flue gas. Per cent of air used for combustion.

This is equal to: $100 \times \dfrac{79.Y}{21.X - 79.Y}$

where Y = the oxygen per cent in the flue gas.
X = the nitrogen per cent in the flue gas.

45 Heat carried away by steam in the flue gas.
Find total steam formed by burning hydrogen and from moisture in coal. Multiply this by the total heat of the steam, minus the total heat of water at air temperature: this difference is equal to — 1082 — t + .48T. where T = temperature of flue gas, deg. fahr. and t = temperature of air.

46 Calculate in the usual way from each flue gas analysis

$$\frac{700 \times 4\, CO_2 \times O_2}{3\,(CO_2 \times CO)}$$ and report the average of the computations.

DISCUSSION

E. H. LOCKWOOD: I am one of the members who has been free to criticize the Code for Testing Low Pressure Heating Boilers as it was originally adopted. I expressed my disapproval in 1918 to Professor Breckenridge, and again in 1919 to Prof. John R. Allen. He told me that the Bureau of Mines had offered similar criticisms which would be presented to the Society. These criticisms have now been published and are in practical agreement with my own ideas. There are four items where I hope to see improvements made in the Code, and I offer the following suggestions on them:

New-Fire Method of Starting.—This method was disapproved by two members of the Committee, John R. Allen and L. P. Breckenridge, both of whom stated that they never used it for starting tests. Many of us agree with these good authorities, and object to the presence of the new-fire method in the Code. It is a poor method and does not deserve approval by the Society. The Committee would do well to recommend its omission.

Efficiency Based on Combustible.—The efficiency of boiler and grate, also known as the over-all efficiency, is the only one of practical value. The second efficiency based on combustible, also called the efficiency of the boiler, is a higher value obtained by crediting the boiler with the unburned fuel found in the ash. The steps in this procedure are to determine the ash in the coal from the analysis, and compare this with the ash weighed during the test. If the ash found in the test is greater than that by analysis, the difference is assumed to be unburned fuel, which is accordingly deducted from the weight fired, thus raising the efficiency. If the ash by analysis and the ash from the test agree in amount, both efficiencies are identical. Experience shows that the weight of ash from a boiler test varies, not only with the unburned fuel it contains, but also with the skill used in cleaning the fire at the beginning and end of the test. Moreover the computer is quite likely to fall into error in making the calculations for the second efficiency, a fact that every teacher knows well.

A timely example is to be found in the January, 1921 issue of the JOURNAL, pp. 4 and 5, where a boiler test is reported in accordance with this Code, containing both efficiencies worked out, and with obvious mistakes in item 47 called "Efficiency of Boiler, per cent." The two objections are that the average computer will make mistakes in his calculations, and that the ash by test is subject to such variations in amount that inconsistent results often arise.

Heat Balance.—Should the Code contain the items: "Losses, grate, per cent"; "Losses, stack gas, per cent"; "Losses, incomplete combustion, per cent"; "Losses, unaccounted for, per cent"? These items may have value for the technical student, but they have little interest for the heating engineer. He is interested in the proportion of the heat that gets into the steam, and knows that the rest is lost. Why not drop the heat balance?

Rating.—The Committee should revise the section on Rating, which is illogical and unsatisfactory. It might be well to recognize frankly that no standard for rating low-pressure boilers has been adopted by the heating profession. It would be proper, of course, for this Society to recommend some rule or standard for rating boilers, and publish the same with its approval.

The proposal of the present Code to adopt the test capacity as the rating of a boiler is meaningless, as it shows a lack of understanding of what rating stands for.

FRANK B. HOWELL: My order of thought parallels Professor Lockwood's in a general sense. It would be a wise thing if the manufacturers of low-pressure sectional house-heating boilers adopted a standard method for rating boilers. It seems to me, however, that the heating engineer is much more interested in knowing exactly what a boiler will do at all percentages of its catalog rating. If one buys a motor or a dynamo, the vendor of such apparatus provides an economic chart which defines exactly what the motor or the generator will do. He does not go into the detail of how many miles of wiring, or the size of the wire, nor does he submit drawings showing the construction, but he provides an economic chart which is a certification of the performance of that piece of apparatus.

If the manufacturers of low-pressure heating boilers were to provide a similar chart, it would aid the heating engineer in his selection of the proper size of boiler. I would like to submit a chart of this character (see Fig. 1) showing the number of consecutive hours the heating boiler will operate without recharging or other attention, burning anthracite, at catalog rating, or at greater or less than catalog rating and the efficiency of the heating boiler when operating at rating or at greater or less than rating. On the bottom of the chart appears the catalog rating in per cent. The right hand vertical legend reads, Interval in Hours the Boiler will Operate without Attention; the left hand legend reads, Boiler Efficiency.

As an example of the use of the chart, follow along the bottom figures to 100 per cent, which is the catalog rating. Follow the vertical ordinate at this point upward to where it intersects the curve entitled, Interval in Hours Boiler will Operate without Attention and at this point follow the horizontal ordinate to the right edge of the chart where is indicated that the boiler, when performing at catalog rating, will maintain the boilers rated output for 8 hr. without attention.

If a boiler will deliver an output equal to catalog rating (100 per cent) for 8 hr. without attention, it is evident that the fuel carrying capacity of the boiler is sufficient to operate continuously at that output rate for 8 hr. without attention.

The curve entitled Boiler Efficiency on the chart would enable the heating engineer to see at a glance the heating boiler manufacturers certified percentage of efficiency at any boiler output, whether at catalog rating or at greater or less than catalog rating.

The heating boiler manufacturer is often requested to specify the evaporative efficiency of a boiler. As the efficiency changes with the

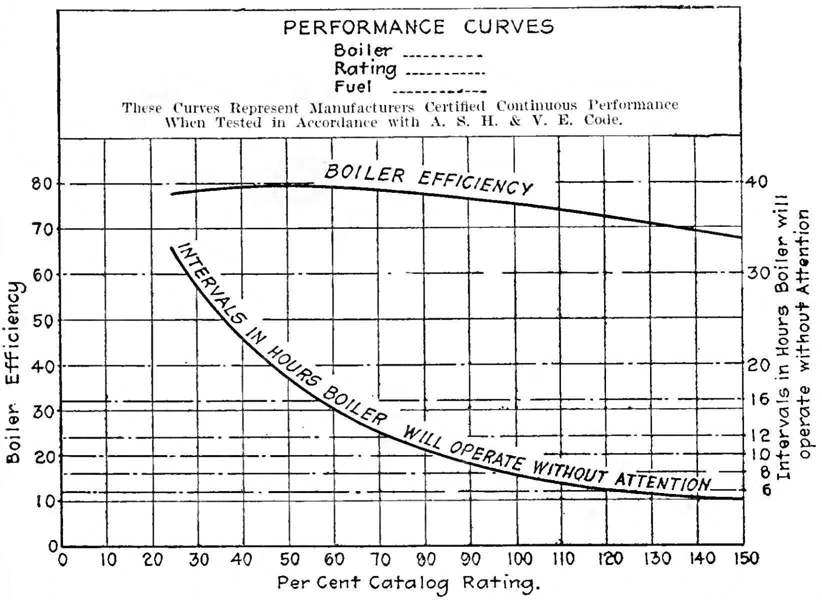

FIG. 1. CHART SHOWING IDEAL PERFORMANCE OF TYPICAL HEATING BOILER.

boiler's output rate, it is not specific to affirm an efficiency without also giving the percentage of catalog rating at which the heating boiler operates when developing the given efficiency. An examination of the curve entitled Boiler Efficiency (Fig. 1) shows the evaporative efficiency at all output rates.

The weather variations during a heating season cause considerable fluctuations in the load, hence the heating boiler is required to develop catalog rating at brief intervals only, operating for perhaps 80 per cent of the winter at between 40 and 60 per cent of its load (not rating). Therefore, an efficiency curve would be helpful to the heating engineer in selecting a boiler that would perform with a high average efficiency all winter.

If heating boiler manufacturers would provide with their boilers upon the purchase an economic chart for the engineer, which would supply information often sought by him now when he is compelled to ascertain the heating surface or the pounds of fuel comprising a charge or in reserve after a boiler has developed its catalog rating for 8 hr. without attention, it would enable the engineer to perceive graphically every material quality of a boiler's performance. The legend at the top of the chart reads: These Curves Represent Manufacturers Certified Continuous Performance When Tested in Accordance with the A. S. H.-V. E. Code.

The A. S. H.-V. E. Code specifies up to 80 per cent as available and 20 per cent for rekindling fuel charge; 20 per cent is not a satisfactory rekindling charge on small-sized boilers when operating continuously. A fuel charge of such proportions enables a single eight hour test to be developed splendidly, but at the end of the eight hours there will not be live fuel enough in the boiler to develop rating a second eight hours, because it takes a couple of hours to build up to rating again after recharging it. A chart will disclose such shortcomings.

There are a number of laboratories throughout the country, especially equipped to test low-pressure heating boilers and I dare say the method of procedure of each is substantially similar, as well as the treatment and the computation of the test data.

It seems to me that the manufacturers could get together readily and standardize a heating boiler test code and show a boiler's varying time without attention outputs, and its coincident percentages of efficiency, on a chart.

If a boiler burns anthracite or bituminous fuel and does not burn the volatile gas, it will certainly be reflected in the efficiency curve. If provisions have been made for burning the volatile, it will show in the efficiency curve. If a boiler manufacturer has made his boiler skimpy, so it will not run 8 hr. at rating and then run another 8 hr. continuously, at rating it will show itself on the curve, Intervals in Hours that Boiler Will Operate Without Attention.

There have been made what the manufacturers know as 6 hr. boilers, that will do their work continuously at rating 6 hr., but they will not develop their rating for 8 hr., because they have not sufficient fuel carrying capacity. No ratings should be established on any heating boiler, small or large, based on a short time test. The test duration should cover several firing and several shaking and fire-cleaning periods.

I would like to go back to the 100 per cent of rating ordinate for a moment, and to the first time-period of 6 hr. If a manufacturer records the performance at rating of a 6 hr. boiler on a chart like this, the curve entitled: Interval of Hours that Boiler will Operate Without Attention, will not be where this one is, but it will cross the 100 per cent of rating at 6 hr. And the heating engineer automatically knows right then and there that it is a 6 hr. boiler.

EDWIN A. MAY: As a member of this Committee it seems to me that we are gradually getting away from the intent of the Code for Testing Low-Pressure Heating Boilers. When the Committee was first appointed to establish a Code for Testing Heating Boilers, it was the idea that the Code could be so simplified that the heating engineer could go out on a job and make a commercial test. Since the first report of the Committee, constructive criticism has been received so that the Code as now presented has gradually grown to the point of making it a laboratory test condition and has had included a great many items which from a practical standpoint are not at all required to establish what the boiler is doing.

I think Professor Lockwood has really misinterpreted the Committee's thought on the rating of a boiler. The real formula given, wherein it states that the rating shall, by this particular Code, be developed by following out certain procedure, was not intended at all to give the idea as to the commercial sales rating of any boiler, but to establish for the benefit of the engineer who was making that original test what the boiler was developing under that particular test in B.t.u. output.

I am rather disappointed at seeing it get into the final technological side of boiler testing, because I think Mr. Howell has expressed very decidedly what has always been the thought of most engineers, that if the boiler manufacturers would only give them something to work on, they would not worry about the testing. But formerly the boiler manufacturers hesitated and kept back information which might just as well have been given out. So the Society felt that it must push some rating so that engineers could go out on a job and make a preliminary simple test and establish for themselves whether the boiler was doing what it ought to do.

P. J. DOUGHERTY: The subject of rating boilers is of vital interest to every member of our Society. Professor Lockwood states that "no standard for rating low-pressure heating boilers has been adopted by the heating profession." I do not agree with him. Our Society in my opinion, is the only recognized authority capable of speaking for the heating profession. Our Society has adopted a Code for Testing Heating Boilers which includes a formula for rating boilers. Furthermore, the late Dean Allen states in the April, 1920 JOURNAL, that our Research Laboratory has endorsed three of the ten standards adopted by this Society, one of the three being: "Report of the Committee on Code for Testing Low-Pressure Heating Boilers," 1919 Revision (TRANSACTIONS VOL. 25, 1919, p. 143).

The Boiler Code endorsed by our Research Laboratory gives the formula for calculating boiler ratings or capacity, and sets forth only three factors: the available fuel, the efficiency or evaporative power, and the length of firing period. Those three factors tell the whole story. The grate area factor used by Professor Lockwood and many engineers and architects is ignored in the Boiler Code rule, and its fallacy is fully set forth in the September, 1920 JOURNAL, p. 608. All boiler designing engineers and boiler manufacturers who maintain their own research departments, use the three factors set forth in our Boiler Code and ignore the grate area and heating surface area factors handed down from high-pressure practice.

It will be noted that the boiler performance curve shown in Fig. 1, contains the three factors set forth in our Boiler Code and ignores the grate area factor. The length of firing period depends upon the type of installation. In residence work with no janitor service the eight-hour firing period is the general rule and should be changed from this competitive eight-hour basis to a ten-hour basis as it should be to suit the requirements in the average home. It is to be hoped that the various chapters of the *Heating and Piping Contractors National Association* will adopt such a rule. The largest types of boilers in charge of janitors are usually rated on a six-hour basis with hard coal. All boilers burning soft coal, because of the nature of the coal, are rated on an hourly basis. The length of firing period is the most important of the three factors in our Code rule.

E. H. LOCKWOOD: I would like to add a final word on the criticism published in the January issue of the JOURNAL, offered by Mr. John Blizard, fuel engineer of the Bureau of Mines. His criticisms reach to almost every paragraph of the Code, and prove that, in spite of revisions by the Committee, plenty of blemishes still remain. In my opinion Mr. Blizard's criticisms are well taken, and express the feelings of many engineers who are competent to discuss the Code. His comments are helpful in many instances, and doubtless will be carefully considered by the Committee. Personally I like Mr. Blizard's new Report Form, offered as a substitute for the present form, but am not optimistic enough to expect its adoption unless considerably modified. Heating engineers will hardly endorse the suggestion to omit the term *foot of radiation* and to substitute for it the new term *1000 B.t.u.* Likewise *equivalent evaporation, from and at 212 deg.* has found so wide adoption in all boiler report forms, that it will be difficult, and perhaps unwise, to drop it. It would be interesting to hear from our members on this subject, as well as from Mr. Blizard.

THE PRESIDENT: Mr. Blizard is not here. Is there any one here who can speak for him on that?

F. B. HOWELL: Relative to this pending question I beg leave to refer the members to Technical Paper 240 of the Department of the Interior, Bureau of Mines, prepared by Rufus L. Strohm, entitled: Boiler and Furnace Testing. This paper defines a very simple and practical method of testing heating boilers, which is fundamentally the same as the methods found in any standard text book. No one appears to emphasize the importance of the "time without attention periods," nor the long duration essential for making a test to develop a heating boiler's rating. The method described in this book differs but little from the method of testing high-pressure apparatus; fundamentally it is the same.

No. 582

REPORT OF COMMITTEE ON STEAM AND RETURN MAIN SIZES

To the Members of the Society:

As a result of the instructions given to your Committee at the Annual Meeting, 1920, to collect data on standard sizes of low-pressure steam and return mains, a questionnaire was sent out to a large number of engineers who are believed to be interested in this question, relative to certain tabulated pipe sizes and there was also included a list of questions for the purpose of bringing out data concerning the various types of systems. This questionnaire was also published in the April issue of the Journal of the Society, p. 1, with a note inviting the assistance of the membership. It also requested the various local Chapters to appoint sub-committees to aid in the work. It was the thought of this Committee that the publication of the tables of pipe sizes and the questionnaire would assist the Chapters in their discussions at meetings. The list of questions submitted is as follows:

1. Have you made any tests or observations to determine the critical velocities at which the down-flow of condensation in steam risers is balanced by the up-flow of steam?
2. Have you collected any field data on the increase in friction in steam mains, due to entrained water and condensation and to the average number of fittings in standard installations, over that of straight pipe and dry steam?
3. Have you any information that might be used in the compilation of a table of vacuum-system return mains for any desired drop vacuum similar to that of Table 5, on steam mains?
4. You are requested to check the portions of the various tables which agree with your present practice, and to submit complete tables of the sizes which you use where your practice differs from these tables.

The tabulations of pipe sizes embrace six distinct types of systems, the number having increased very much since this subject was previously investigated by the Society in 1906, so that it has been thought necessary to prepare a separate table for each type of system.

The response to this questionnaire, owing to lack of individual research work by engineers and engineering contractors, has been so limited up to the present date as to be of little use to the Committee, and after mature deliberation, your Committee has decided to recommend that the subject be submitted to the Research Department of the Society for formal investigation.

You are, therefore, urged to recommend to the sub-committee on subjects of the Committee on Research, that definite instructions be given for research work to be carried out along this line with a view to obtaining the information which your Committee has hoped to be able to obtain, but cannot bring out in any other way. For the information of the Committee on Research, it should be stated that several college and university laboratories have carried out research work along these lines in a limited way and several others have indicated their willingness to cooperate with the Research Department of the Society to this end, if it is desired.

With a view to rendering this suggestion for additional research work constructive, a proposed outline for research in steam and return main sizes is herewith presented, in the hope that the research work will be divided into four general classifications as there indicated, supplemented by such discussion as may be had upon it at the Annual Meeting. The proposed outline is appended to this report.

Respectfully submitted,

(Signed) JAMES A. DONNELLY, *Chairman.*

PROPOSED OUTLINE OF RESEARCH IN STEAM AND RETURN MAIN SIZES

I. Classes of Steam Mains:

1. Distribution mains, or mains conveying steam to or through buildings.

NOTE: The only consideration in distribution mains is the available or commercially advisable drop in pressure, but provision must be made at points of connection to service mains for proper withdrawal of condensation.

2. Service mains and branch piping from distribution mains to radiation.

NOTE: First consideration in these is the critical velocities*, or those which allow of sufficient separation of water and steam so that water hammer or defective circulation will not occur. Second, allowable drop in pressure so that water will return properly to the boiler.

a. Vertical risers:

1. One pipe; 2. Two pipe; 3. Up-feed for down feed systems; 4. Down-feed risers; 5. Connecting risers at top.

b. Horizontal offsets in risers:

1. One pipe; 2. Two pipe.

c. Horizontal radiator connections pitching back:

1. One pipe; 2. Two pipe; 3. Those serving more than one radiator; 4. Vertical portion—one-pipe and two-pipe.

d. Effects of length of radiator; length of connection, amount of pitch and number of elbows on pipe size in one-pipe radiator connections.

e. Radiator connections draining into two-pipe radiators.

* It is possible by using velocities which are well above the critical velocities to secure very fair results except when raising or lowering pressure or when some radiators are shut off in mild weather, and this is apparently the practice in Europe. In this country, we use much larger pipe sizes and keep below the critical velocities at all times.

f. Riser connections:
1. Pitching with steam; 2. Pitching against steam; 3. One-pipe; 4. Two-pipe.

II. Inlet Valve Sizes:
1. One pipe; 2. Two pipe; 3. Two pipe air return system; 4. Two pipe vacuum systems.

III. Return Mains:
1. Wet returns:
a. One pipe.
b. Two pipe.
2. Dry returns:
a. One pipe system.
b. Two pipe system.
c. Two pipe air return system.
d. Two pipe vacuum system.
3. Return risers:
a. Two pipe gravity.
b. Two pipe air return gravity.
1. To wet main return.
2. To dry main return.
c. Two pipe vacuum return system.
4. Return radiator connections:
a. Two pipe gravity.
b. Two pipe air return gravity.
c. Two pipe vacuum return system.
5. Return valve sizes:
a. Two pipe gravity.
b. Two pipe air return gravity.
c. Two pipe vacuum return system.

IV. Drips:
1. From distribution mains:
a. High pressure.
b. Low pressure.
1. On the line.
2. At connection to servce mains.
3. Necessity of eccentric fittings?
2. From service mains:
a. Gravity.
b. Gravity through traps or loop seals.
c. Vacuum return through traps or loop seals.
3. Riser drips:
a. One pipe system.
b. Two pipe system.
c. Two pipe air return system.
d. Gravity through traps or loop seals.
e. Vacuum return through traps or loop seals.

DISCUSSION

THE CHAIRMAN (J. A. Donnelly): Our present piping sizes have been developed from our experiences with priming boilers and wet steam. We do not know what pipe sizes we could use if we always had perfectly dry steam. We do know, however, that where we use exhaust steam or live steam reduced to low pressure, that we have less piping troubles and can use somewhat smaller pipes than when the steam is supplied from low-pressure boilers. This, we have concluded, is due to the effect of boiler priming.

After I had this report published, Mr. Timmis suggested an addition to it which had to do with the location of thermostatic valves sufficiently far from the drip points so they would work well. At first glance, that may not have anything to do with pipe sizes, but it does. It seriously interferes with the proper dripping of the steam main if it is located too close to the drip point.

E. A. MAY: I think that this Committee has done very good work and I am glad Mr. Donnelly made the distinction that this Committee was appointed to collect data. I think if that Committee has collected any data whatsoever then this Society is entitled to that collection. And the only thought I have in daring to talk about pipe sizes is that we have gone along for years and everybody has religiously designed his supply line and then consistently upset the balance by guess. It seems to me that we can arrive at some method of establishing return line sizes which will be consistent with the supply line. Whether the principle is right or wrong is to be established by the suggestions of the Committee and by actual research. If a consistent relationship can be established which will comply with the majority of the engineers' practice, then I believe such a system or method can be printed, not necessarily endorsed by this Society logically.

With this in view I desire to submit here a suggested solution of the problem for your consideration:

PROPOSED RULE FOR PIPE SIZES

The carrying capacity of pipes as applied to vapor and vacuum heating systems is governed by the difference in pressure between the two ends of the pipe or *pressure drop*, the density or weight per cu. ft. of steam or water, and the frictional resistance of the pipe. Various formulae have been advanced showing the relation of these factors to the final result, the most generally accepted being the Babcock formula which is as follows:

$$W = 87\sqrt{\frac{P.C.d^5}{L\left(1 + \frac{3.6}{d}\right)}} \qquad (1)$$

in which W = weight of steam passing through pipe in lb. per min.;
P = difference in pressure between the two ends of pipe in lb.;
C = density of steam or weight per cu. ft.;
D = actual inside diameter of pipe (in.);
L = length of pipe in ft.;

This formula modified to express result in sq. ft. of radiation per hour or equivalent would read:

$$R = 17400 \sqrt{\frac{P.C.d^6}{L(d+3.6)}} \qquad (2)$$

in which R = sq. ft. of radiation based on condensing factor of 0.3 lb. per sq. ft. per hour.

In the above formula the coefficient of friction for steam is taken as 0.0027. In using this formula for determining return-pipe sizes where water and air are carried, a coefficient of 0.005 has been used, which modifies the formula to read:

$$R = 11600 \sqrt{\frac{P.C.d^6}{L(d+3.6)}} \qquad (3)$$

In the selection of pipe sizes for vapor heating systems, it is essential on account of the extremely low pressures employed, that the frictional losses in the system should not exceed a few ounces.

Supply Main: The supply main and riser sizes given in the following tables for vapor heating are based on a pressure drop of 1 oz. in 100 ft. of pipe or its equivalent. (Due allowance should be made for ells, and valves in line of flow.) It is assumed that each square foot of radiation or its equivalent, including supply piping, will condense 0.3 lb. steam per sq. ft. per hour.

Dry Return: The dry return in a vapor system carries all of the condensation from the radiators to the boiler, as well as the air from the system to a point where it is expelled. In compiling these tables it is assumed that the radiators are to be equipped with a thermostatic trap which allows no steam to pass into the return line.

The density of the air and water flowing through the return lines per hour has been determined as follows:

One sq. ft. of average radiation plus an allowance for piping will contain approximately 2 lb. of water or 0.032 cu. ft. of volume. The condensation from 1 sq. ft. of radiation at 0.3 lb. at a density of 60 lb. per cu. ft. will give a volume of 0.005 cu. ft. of water per sq. ft. per hour. As a factor of safety the extreme assumption is made that 0.032 cu. ft. of air as well as 0.005 cu. ft. of water will pass through the dry return per hour per sq. ft. of radiation which is in the ratio of 6.4 volumes of air to one of water giving a density or weight per cu. ft. of mixture of 8.17 lb.

Wet Return: In determining the size of wet returns we have taken into consideration the fact that the pipe is entirely filled with water and that no provision must be made for carrying steam or air. The temperature of the return water has been taken at approximately 200 deg. and a density of 60 lb. per cu. ft., which factor has been used in the formula. The frictional coefficient of water has been taken into account in establishing the constant of 11,600, as used in equation (3).

Risers: In establishing the sizes of risers, we have taken into consideration the fact that the condensation in the feed riser is flowing in an opposite direction to the steam and in the small sizes the velocity has been taken at from 10 to 15 ft. per sec., which experience has indicated is a safe factor in preventing any interference to the flow of condensation. In larger sizes of pipes the allowable velocity is in excess of 15 ft., inasmuch as the velocity decreases from the center towards the surface of the pipe and the actual velocity at the surface does not interfere with the downward flow of condensation. Therefore, the formula used for establishing main supply is used on sizes larger than 1½ in. In the return risers as air and water only are to be handled, and both flow in the same direction, the return riser sizes have been established by use of the same formula as that used for the dry return. (Equation 3.)

In the down-feed risers the same formula is used as for main supply, as condensation and steam are flowing in the same direction.

Branch Connections: For branch connections where the condensation is flowing with the steam, the same capacities can be used as for steam mains, but where the condensation flows against the steam in a horizontal pipe, it is recommended that one size larger pipe be used than for the riser, so that the velocity of flow may be kept to the proper minimum.

Supply Radiator Arms: The sizes of supply radiator arms have been established on safe velocities which will not interfere with the flow of condensation.

Return Radiator Arms: The sizes of return radiator arms are based on the same formula as dry return—as only air and water are to be handled.

Drips: The sizes of drip pipes are based on wet return data, given in equation (3).

In addition to the composite Tables 1 and 2 which furnish pipe size data for average conditions met with in vapor and vacuum installations, we are presenting Tables 3 and 4 giving the capacities of steam mains and return mains for vacuum heating work at varying losses in pressure. We believe these tables will be found to be very useful to the designing engineer.

In order to use the tables, it is necessary to proceed as follows:

1. Determine whether a vapor or vacuum system is best suited to the conditions surrounding the installation; after deciding this, determine the source of steam supply and the initial pressure to be carried;
2. Determine the total amount of radiation and points at which branches are taken from the main;
3. Determine the total distance from the source of steam supply to the last branch supply on the main, to which should be added 50 per cent additional length to cover friction in valves and ells;
4. The allowable pressure drop in the supply main, which for good practice may be taken as one-half of the initial pressure;
5. Multiply the total pressure drop in ounces by 100 and divide by the total length of the main in feet, which will give the pressure drop in ounces per 100 ft. of length. After this is determined, select proper column in Tables 3 or 4 for designing steam mains and return main.

EXAMPLES

Example 1: It is decided that a vapor system is to be used. Steam is to be supplied from a low pressure boiler at an initial pressure of 6 oz., the length of supply main from boiler to last branch connection being 200 ft. (to which should be added 100 ft. to provide for friction in valves, ells, etc.). Total amount of radiation 2,000 sq. ft. equally distributed in length of main.

The allowable pressure drop in supply main, taken as one-half of the initial pressure would be 3 oz., leaving available for loss in return lines 3 oz. Multiply 3 oz. by 100 and divide by 300 (total equivalent length of run which will give an allowable pressure drop of 1 oz. in 100 ft. run). By referring to 1 oz. column in Table 3 we find 2,000 sq. ft. opposite 4 in. pipe. This size should be used from the source of supply to a point where the radiation to be supplied is reduced to the amount opposite the next small size of pipe, which in this case is 1,436 sq. ft., opposite 3½ in. pipe. This size would be run to a point where the radiation has been reduced to 961 sq. ft., where it would be reduced to 3 in. A further reduction in size would be made to 2½ in. when there remained to be supplied 525 sq. ft. This size should be run to the end of the main, as good practice and experience have shown that a main starting larger than 2½ in. should not end smaller than 2½ in.

In a similar manner, the dry return lines would be proportioned, using the capacities given under the 1 oz. column in Table 4, and, for such return lines as may be below the water line of the boiler, carrying water only, using Table 5.

Example 2: A vacuum system to be used; the initial pressure, 3 lb.; equivalent length of main 600 ft.; 10,500 sq. ft. of radiation.

This initial pressure would permit a drop in pressure in supply main of 1½ lb., or 18 oz., and a like drop in pressure in the return lines. Multiply 18 oz. by 100 and divide by 600 (length of run) would give a drop in pressure of 3 oz. in 100 ft. By referring to 3 oz. column in Tables 3 and 4, the size of supply main would be found to be 6 in. for 10,500 sq. ft., and would be decreased to 5 in. for 6,511 sq. ft. —4 in. for 3,520 sq. ft.—3½ in. for 2,488 sq. ft.—3 in. for 1,665 sq. ft.—2½ in. for

910 sq. ft. The return main 3 in. for 10,500 sq. ft., 2½ in. for 8,765 sq. ft., 2 in. for 5,323 sq. ft. 1½ in. for 2,623 sq. ft., 1¼ in. for 1,690 sq. ft., etc.

With a drop in pressure of 1½ lb. in total length of return line, a vacuum pump maintaining a vacuum of from 3 to 4 in. at the pump would give satisfactory results.

In designing piping sizes and lay out, after determining amount of radiation and points at which branches and risers are taken from mains, the proper sizes may be determined by starting at the farthest point and working toward the source of supply. This method is used by many heating engineers.

TABLE 1. CAPACITIES (SQ. FT. DIRECT RADIATION OR EQUIVALENT) OF PIPING FOR VAPOR SYSTEMS

Initial pressure 6 oz.—1 oz. loss in 100 ft. equivalent run.

Pipe Size	Steam Main	Risers Upfeed and Downfeed	Radiator Supply Arm	Dry Return and Return Risers	Wet Return
¾				220	
1		45	45	444	1,200
1¼		100	100	976	2,645
1½		155	155	1,516	4,107
2	315	315		3,074	8,333
2½	518	518		5,064	13,720
3	949	949		9,269	25,120
3½	1,417	1,417		13,847	37,524
4	2,004	2,004		19,586	53,077
5	3,708	3,708		36,240	
6	6,087				
8	12,670				
10	23,121				

TABLE 2. CAPACITIES (SQ. FT. DIRECT RADIATION OR EQUIVALENT) OF PIPING FOR VACUUM SYSTEMS

Initial pressure 21 lb.—4 oz. drop in 100 ft. equivalent run.

Pipe Size	Supply Main	Risers Upfeed	Risers Downfeed	Return Main	Return Riser	Radiator Supply Arm
¾			48	431	431	
1			96	887	887	
1¼	212	125	212	1,951	1,951	125
1½	329	225	329	3,029	3,029	125
2	668	668	668	6,146	6,146	
2½	1,100	1,100	1,100	10,120	10,120	
3	2,014	2,014	2,014	18,530	18,530	
3½	3,008	3,008	3,008	27,680	27,680	
4	4,255	4,255	4,255	39,150	39,150	
5	7,873			72,430	72,430	
6	12,923					
8	26,900					
10	49,090					

TABLE 3. CAPACITIES (SQ. FT. DIRECT RADIATION OR EQUIVALENT) OF PIPING—SUPPLY MAINS

Varying pressure drops—100 foot run.

Pipe Size	1 Oz.	2 Oz.	3 Oz.	4 Oz.	6 Oz.	8 Oz.	10 Oz.
1	46	65	80	92	114	123	148
1¼	101	143	175	203	252	291	325
1½	157	222	272	314	391	452	505
2	319	451	553	638	793	916	1,024
2½	525	743	910	1,051	1,307	1,509	1,687
3	961	1,360	1,665	1,923	2,392	2,761	3,087
3½	1,436	2,032	2,488	2,873	3,572	4,125	4,612
4	2032	2,874	3,520	4,064	5,053	5,835	6,523
5	3,760	5,317	6,511	7,519	9,349	10,795	12,069
6	6,170	8,727	10,685	12,340	15,346	17,720	19,811
8	12,845	18,167	22 247	25,690	31,946	36,890	41,240
10	23,440	33,150	40,600	46,880	58,300	67,310	75,250

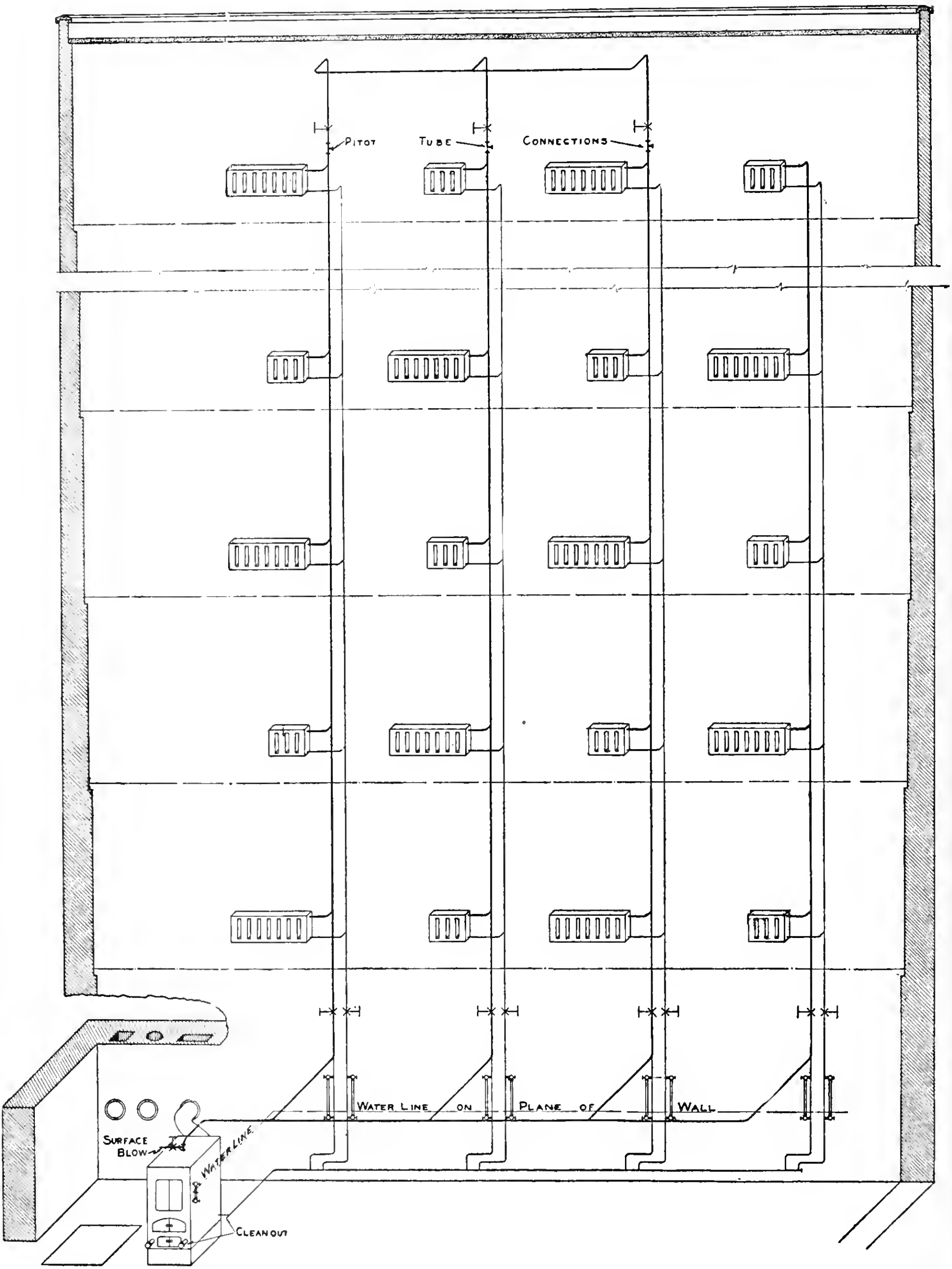

FIG. 1. CHART SHOWING PROPOSED METHOD OF INVESTIGATING THE PERFORMANCES OF STEAM AND RETURN MAINS IN A BUILDING.

TABLE 4. CAPACITIES (SQ. FT. DIRECT RADIATION OR EQUIVALENT) OF PIPING—DRY RETURN

Varying pressure drops—100 foot run.

Pipe Size	1 Oz.	2 Oz.	3 Oz.	4 Oz.	6 Oz.	8 Oz.	10 Oz.
¾	220	312	382	440	540	623	697
1	443	627	768	887	1,086	1,198	1,402
1¼	975	1,380	1,690	1,951	2,390	2,759	3,085
1½	1,515	2,142	2,623	3,029	3,710	4,284	4,789
2	3,073	4,346	5,323	6,146	7,527	8,692	9,718
2½	5,061	7,157	8,765	10,120	12,396	14,314	16,004
3	9,264	13,101	16,046	18,530	22,691	26,202	29,295
3½	13,838	19,570	23,969	27,680	33,896	39,140	43,761
4	19,575	27,683	33,904	39,150	47,947	55,365	61,900
5	36,216	51,217	62,727	72,430	88,708	102,435	114,525

TABLE 5. CAPACITIES (SQ. FT. DIRECT RADIATION OR EQUIVALENT) OF PIPING—WET RETURNS

Varying pressure drops—100 foot run.

Pipe Size	1 Oz.	2 Oz.	3 Oz.
1	1,203	1,700	2,084
1¼	2,645	3,740	4,581
1½	4,107	5,807	7,113
2	8,333	11,783	14,433
2½	13,720	19,400	23,771
3	25,120	35,520	43,508
3½	37,524	53,060	64,991
4	53,077	75,050	91,929

JAMES A. DONNELLY: I have been puzzled for some time as to how we could overload and underload the steam risers of a building without really giving trouble; how we could research on rather a large scale. It is rather expensive to put up a building and devote it to research; and yet research without something of considerable size might be a trifling laboratory experiment. I finally devised two ways of doing it. One might be to put radiators of different sizes on the risers and switch them, as 40 and 80 ft. radiators, and switch the 80 and take the 40 and so underload and overload. That would be in a building where the floor was practically open, without partitions. Such a building might be available. The Detroit Edison Company some time ago suggested that they would like to do experimental work, especially in return pipe sizes, in a basement, where in a building of their own they could take a little more chance than other people could, where they would have mechanics and pipes available.

Another method, which is simpler still, is to devise a building with 50 or 100 per cent more radiation than it needs, and in that way overload the risers. Then by putting the key valves on and shutting off the radiators and opening them, we could devise greater and less loads.

In connection with this same thing it is very easy to provide water columns both in the drip and return line, so as to read the pressure in the drip and return mains. Provide a double space for boilers with several chimneys, perhaps of different forms, characteristic chimneys, made with round, square or oblong chimney flues.

Then we might have the possibility of testing a plant after it is put in, with all the fouling of grease and scale and sand and cutting compounds, etc. I think we could get the drop in pressure due to boiler priming and have a method of applying indicating instruments to the supply mains, so when that boiler again began to prime, we could apply the proper remedy.

As the riser gets overloaded and the water dances up and down with the steam, I have often found it on the top of the riser by the chart of the recording gauge. We want to get recording steam gauges more sensitive, so they will record very slight variation in pressure. I have one in use now that shows the suspension of water in the riser of our building in New York. I had some charts from a building that I took about five or six years ago that had a variation of 1 or 1½ lb. difference in pressure, where water would collect and be thrown out. I will send that to the Research Laboratory.

There is a remedy for some of these things you have in both one and two pipe systems. I suggest connecting both risers by a little auxiliary steam line. That gives many advantages, down-feed without an up-feed riser, and without having a series of large mains marring the top floor in many cases. These connecting pipes would be one size larger than the steam riser, as the water would not collect in all the risers at the same time.

Those were about the applications I had in mind when I drew this to illustrate some of these methods.

L. A. HARDING: There is one question I would like to ask the members of the Society, why was a pressure of 1 oz. per 100 ft. of run originally chosen as a standard for designing low-pressure steam lines? As a matter of historical interest, I think the members of the Committee would like to know how that came about originally.

A. A. ADLER: The earliest tables I have any recollection of were Wolf's tables, and it so happens that those old tables just about figure out on that basis. For the last ten years we have been using the Babcock formula for the flow of steam for designing low-pressure steam mains, with a pressure loss of 1 oz. in 100 ft. of run.

The present choice of suitable pressure drop seems to be arbitrary. The real engineering problem lies in determining the permissible drop both from the viewpoints of the economics involved, and the conditions which lead to satisfactory steam circulation at any and all loads on the heating system.

For the first of these viewpoints, consider the delivery of a given quantity of steam through a large or a small pipe. In the case of the large pipe the drop in pressure is due to a large condensation and a small drop due to friction. In the case of the small pipe the condensation will be small and the friction drop will be large. Here a balance can be struck since the same drop in pressure may be obtained in one of two ways.

Some years ago I attempted the solution of this problem but gave it up. Since then I recommended this to a student as a thesis problem. If he solves it I will be pleased to transmit the results to the Society.

The second part of the problem, i. e., the determination of whether the economic drop permits good steam circulation may then be checked up with satisfactory practice.

E. A. May: I move that the recommendation of the Committee be complied with and this subject be referred for formal investigation and development to the Research Laboratory of the Society.

The motion was seconded and carried.

J. A. Donnelly: Professor Harding has asked for some information from some of the members, and perhaps I can give it in just a moment. It was concerning the origin of the standard drop of 1 oz. to 100 ft. It was first suggested in 1905. In the January meeting of that year I published a table of the drop of pressure in pipes, which was about the first one published. I think Briggs had gotten one out based on one-quarter or 2 lb. condensation per sq. ft. of radiation and quite a high drop. A table gotten out by Alfred R. Wolf for 2 lb. and 5 lb. pressure had been published, I think, by the Franklin Institute of Philadelphia, and the drop was not stated.

I went into Mr. Wolf's office and saw Mr. Ohmes and he said one of their men had worked out that table, at a drop of 2 per cent of gauge pressure.

One other phase and I am through. In visiting New York University with Dean Allen and in compiling some lists with him which I do not think ever got into letter form or into print, the subject of pipe sizes was referred to the New York University, and I think they laid out a plan with which they expect to heat their new buildings and are going to make some study of pipe sizes.

Konrad Meier (written): Having made a special study of the subject, the writer may be permitted to discuss the questionnaire issued by the Chairman of the Committee, J. A. Donnelly in the April Journal of the Society and his report of progress presented at the Semi-Annual Meeting. Reference is made to the four questions, although these are answered by comment rather than by the desired collection of data.

1. The critical velocity, as referring to the flow of steam, is a somewhat indefinite term, which is not analogous to the phenomena going under that name for the flow of water. It is apparently intended to designate a rate of steam flow at which irregularities and noise begin to make trouble and any calculation becomes problematical. This rate, in reality would appear to depend more on the relative amount of water flowing against the steam, which is extremely variable while the attainable velocity of the latter is in itself governed by a number of other factors. Even if laboratory experiments could cover all combinations of diameter, length, pitch, local resistances, drip arrangements and rates of heat loss on the run, in practice we would still be in doubt about the initial condition of steam and the temporary increase of flow in reheating. The excess of condensation during that period, which often causes difficulties, is necessarily an uncertain quantity, since it depends not in the least upon the generating capacity of the boiler, which in turn is affected by the draught power, the weather and the operation of the plant. What Mr. Donnelly has in mind is probably the establishment of safe limits for the velocities to be assumed. But the particular conditions to which these will refer, would have to be stated and considered in practice. The

safer way to prevent critical velocities, which must be our aim, would seem to lie in the engineering policy of eliminating the uncertain factors. In this case it means, that we should prevent the undue accumulation of water by proper and ample drainage, while at the same time providing for an easy flow of steam on the principle of an even and moderate pressure drop. This pressure loss between boiler and heating surface is, of course, independent of the working pressure and merely an assumed basis for equalization. If this be accomplished by appropriate sizing of pipes for equal friction and resistance head rather than by artificial adjustments, throttling or reliance on excess pressure, such equalization will not only aid the even flow of steam and the expulsion of the air, but it will prevent also short circuits to returns, which often hold up the condensate from the far end of the line. It assures a steady water line and smooth operation.

Any calculation, however, must be based above all on the true quantities of steam to be carried. The tables given by various authorities, including those presented by Mr. Donnelly, do not take into account all of the essential factors and should serve only for approximation, such as may be required for estimating. They cannot be relied upon for final calculation, because:

a. They do not allow for the variation in pipe capacity due to working pressure or vacuum.

b. They do not remind of the varying rate of condensation in different kinds of heating surface.

c. They do not induce correction for the influence on condensation of room temperature and location.

d. They do not lead to the proper consideration of the heat losses within the steam piping, which is a most variable factor.

e. They do not take into account the condensation within the dry returns, which sometimes materially increases the volume of steam to be carried.

f. They do not allow for the greatly varying length of main and branches.

g. They do not allow for the greatly varying resistance of fittings and other local obstructions.

Altogether they leave much room for errors and discrepancies, that will explain many difficulties experienced in steam circulation, including those from excessive velocities or volumes, which are often grossly underestimated.

2. While it is difficult in laboratories to obtain steady conditions and reliable figures on the friction of dry steam, it seems hopeless to gather field data of any value on the increase of friction due to entrained water and condensate.

Water is entrained in consequence of priming, high water line and small steam space in the boiler and excessive velocities at outlets. It is not only practicable to avoid these causes, but surely more satisfactory. Prevention is better than cure.

If the average number of fittings were known, it would only lead again to assumptions instead of the proper consideration of this factor, which is about as easy, while being the safer basis for calculation.

3. The drop of pressure or vacuum within a system of return piping depends on the amounts of steam, water and air to be handled, which are most variable. For simple gravity systems this drop is generally a small fraction of the pressure loss within the steam lines, especially with wet returns. In vacuum systems the situation changes in so far as returns are dry, additional amounts of steam are condensed within the piping and other apparatus and the volume per lb. of steam becomes greater. When the flow is in the same direction, the steam carried in the dry portions practically determines the size of the returns, the water only becoming a factor towards the lower end of the line and generally calling for a smaller size than the steam carried at the upper end.

The customary sizes of returns are the result of experience with steam mains, which were, on the whole, smaller than desirable and with imperfectly equalized pipe systems. They allow for more or less irregularity of flow, especially in dry return mains. These irregularities, which may become disturbing, are practically avoided, if we remember, that the flow in the returns is largely governed by the pressure conditions in the steam lines where they connect with the returns. Excess pressure at the near branches will result in short circuits and cause the flow of steam against water from the far end, possibly under sudden condensation and noisy disturbance. The key to the proper action of a *return* system therefore really lies in the accurate proportion and liberal size of the *steam* piping, but in so far as the returns must carry steam, they should be considered as part of the latter and the extra volumes involved must be included in the calculation. In theory, the resistances should be equalized down to the water line. In practice, it will suffice to figure on equal pressure loss up to the radiator, but then we must see to it, that the further loss from here to the water line or in the return main becomes negligible. Thus the friction in a bare return riser of considerable length should always be verified and reduced, if liable to upset the equalization. The same applies to bare sections of return mains and care should be taken in such cases, that the steam enters from the upper end. In other words, the steam line leading to it, must be of sufficient size to carry the corresponding extra volume.

These principles apply to all systems of piping and may properly be guiding the calculation of long returns under vacuum, where the increased volumes are a considerable factor. Just as for the steam piping, the customary tables should therefore be used only for approximate sizing, to be verified by the correct calculation on the basis of equal pressure loss in connection with the steam lines.

4. The Tables 1 to 4 presented by Mr. Donnelly for the steam mains and risers in which the condensate flows in the same direction have already been checked, so to speak, by the foregoing criticism. It remains to see, whether they are consistent and safe as approximations. It is stated, that they are calculated for a uniform pressure drop of 1 oz. in 100 ft., but the formula used is not mentioned. Nor is it clear whether allowance is included for fittings. However, the figures indicate a decidedly higher rate of friction for the larger sizes with velocities up to and over 100 ft. per second at which the local resistances are also becoming relatively

higher. At such rates of flow, often materially exceeded in reheating, it is not surprising when water is entrained when there is the least tendency to priming or a high water line. Moreover, such rapid flow cannot be conducive to good distribution, if only because of the uncertain resistance of branch tees and the excessive loss for the far end of the line, which requires undesirably small branches at the near end or adjustment by throttling. The rational calculation on the basis of a moderate and equal pressure drop for all branch circuits will show when smaller sizes are permissible. Generally, it does call for larger mains, but these are desirable in most cases for the reasons stated. Furthermore, they will also make the apparatus less sensitive to disturbances from erratic operation, through shutting off and turning on, as well as against incidental alterations of the plans. Finally, the easy and noiseless circulation will permit quicker reheating and provide better service in general. Hence it would seem advisable to reduce the amount of radiation to be carried by the larger pipes whether the tables are used for approximation or not, and to substitute B.t.u. for sq. ft. or lb. of steam as a more definite basis.

Table 5 evidently is intended for long distance runs, but it is indefinite in regard to the working pressure. The assumption, that fittings and entrained water double the friction is liable to mislead. For long runs, the pressure loss should, in any event, be calculated from some recognized formula or chart under proper consideration of all the factors. The high pressure side of Table 6 seems also based on very rough approximations. It will nearly always pay to figure according to actual pressure and length of run.

In his report of progress Mr. Donnelly recognizes, that what he calls the standard practice of today, could not be a fixed and unchanging guide, but nevertheless he believes, that revision in the light of research would make it safer and discourage material variations. However, the question may well be raised, whether we ought to discourage variations from such approximate tables and thereby encourage the indifferent working method of turning out plans in manufacturing fashion, or if we should, in view of the diversity of the problems, favor their treatment by sound engineering, that is, by systematic and individual calculation. The latter method means the consideration of all the main factors entering into a particular case. It will lead to the study of the mechanical problems involved and of the functions of a pipe system as carrier or distributor, which will then find adequate consideration and expression. It will further bring about good designing, which must go hand in hand with intelligent calculation, recognize and eliminate the uncertain factors and induce precautionary measures for installation, testing and operation, which help to permanent success.

The scientific engineering way can be simplified to suit the stage of the work or its character. It is entirely practicable, but in the last analysis the preference should be given to the method which will assure the best results with the least waste in time, energy and material, and which will make for progress in the art. To that end it seems essential, that we should always know what we are doing, but in this particular subject research today seems to be ahead of its practical application.

No. 583

PROGRESS REPORT OF COMMITTEE ON FURNACE HEATING

PROGRESS reports are herewith submitted along two divisions of the furnace investigations i. e., the Committee on Testing, Prof. A. C. Willard, Chairman, and the Committee on Design, Prof. R. W. Noland, Chairman.

The report from Professor Willard which follows, represents some of the conclusions arrived at from the admirable work being carried on at the University of Illinois and although it has not as yet been submitted to the Sub-Committee of this Society because of lack of time, it proposes a concise and workable summary of procedure which will eventually be in form for final adoption.

The report to me from Professor Noland shows only intangible results at this time. The work of this Committee depends in large measure upon scientific facts obtained from tests and upon actual home working conditions. Data such as those being furnished by Professor Willard are making the problems of design more simple and workable. In addition to this material the Committee has in progress a survey of the local domestic furnace field, which probably represents a fair average of such work, and a study of the good and bad points in design as actually met with in practice. This, in connection with the more scientific data obtained in the laboratories will provide the necessary facts for future recommendations. Suggestions from the members of this Society on any and all points relating to design are solicited.

J. D. Hoffman, *General Chairman.*

Proposed Furnace Testing Codes for Both Pipeless and Piped Furnace Systems as Developed in the Warm Air Furnace Research Work at the University of Illinois

By A. C. Willard[1] and A. P. Kratz[2]

I. THE PIPELESS CODE

Tests under this code may be of two general kinds, depending on character of fuel used. For purposes of comparing similar furnaces, or for most changes in design, it has been found satisfactory to use hard coal and run 8 hour tests. With hard coal, tests are started with clean grates and ash pit with an initial firing of a charge of wood equal to 10 per cent of the full coal charge. The weight of coal to be fired is equal to the fire-pot capacity to bottom of fire door. The wood fire is allowed to burn 15 minutes and then one-third of coal charge is fired. In 15

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[2] Research Assistant Professor.
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minutes more, the second third of the coal charge is fired, and the last third after the expiration of another 15 minutes. The test starts when the first coal is fired, and it will be apparent that all the coal is fired within 30 minutes from the starting of the test.

At the end of 8 hours the fire is completely quenched with water and the refuse dried and weighed. The equivalent coal value of the refuse is then calculated and subtracted from the original weight of coal fired.

When soft coal is used a fire bed is gradually built up for from 5 to 8 hours, and the grates are always shaken and ash pit cleaned just prior to starting the test, with the fire surface practically level with bottom of fire door. The ash pit is then cleaned and the actual test starts immediately. Careful record must be kept

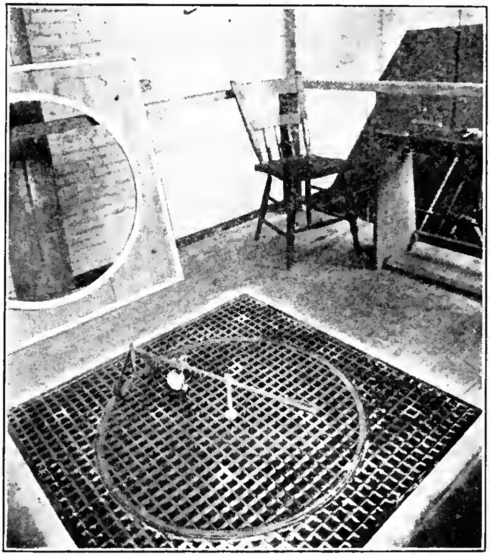

FIG. 1. VIEW OF REGISTER AND SPECIAL TRAVERSING EQUIPMENT WITH ANEMOMETER.

of amount of coal, and length of time which elapsed after firing last coal and shaking grates. The test is then run for not less than 36 hours. Exactly the same procedure is followed in firing and shaking grates at the end of test as was practised when the test started. Samples of coal and ash are taken and analyzed.

All tests are so controlled as to require not more than 0.2 in. of water draft pressure at any time, *and the temperature rise of the air passing through the furnace is maintained absolutely constant throughout the entire test.* Special temperature indicating apparatus is absolutely essential in maintaining a constant temperature rise. In no case, will mercury thermometers be satisfactory for measuring outlet temperatures, because of the serious radiation effects on such thermometers.

The amount of air passing through the furnace is determined by a special traversing equipment (Fig. 1) using an anemometer as an index. The anemometer is then calibrated under exactly the same conditions of air velocity, temperature and register resistance as existed in the actual test.

The observed data and calculated results are then reported on the Data and Result Sheets for Pipeless-Furnace Test given below. Fig. 2 shows the general set-up of the equipment for a pipeless furnace test, and also serves as a graphical result sheet.

II. THE PIPED FURNACE CODE

Tests under this code are made in an exactly similar manner to that already described for pipeless furnaces. The special arrangement of an equivalent piped-furnace testing plant is shown in detail in Fig. 3.

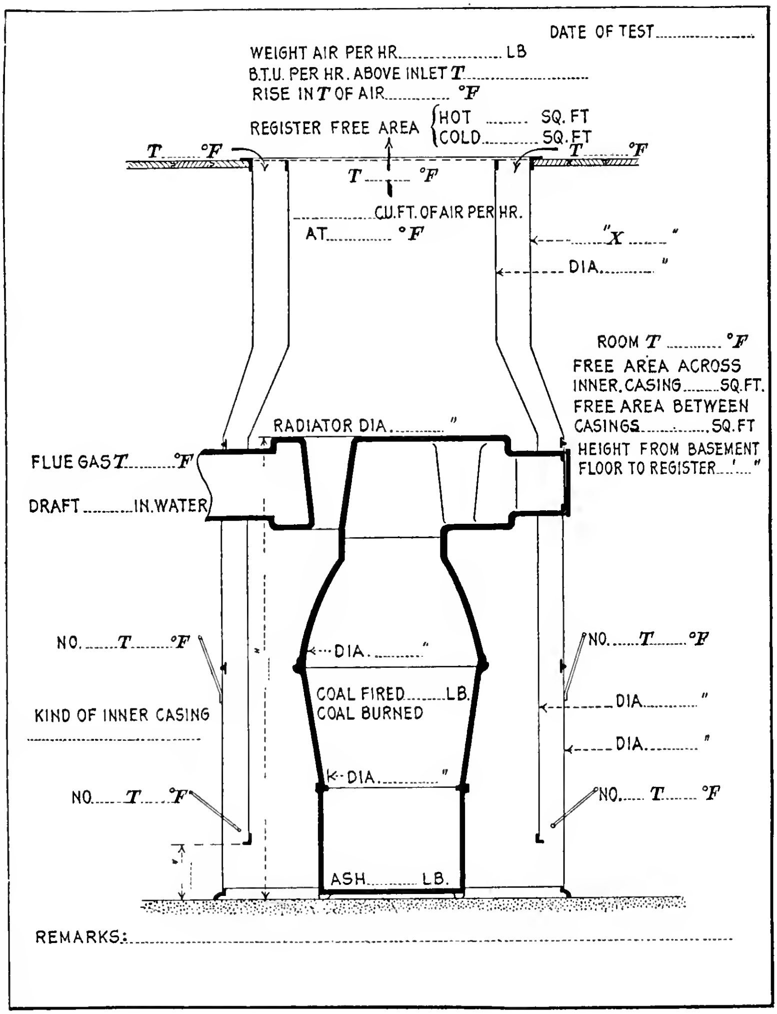

FIG. 2. GRAPHICAL CHART FOR PIPELESS FURNACE TESTING PLANT.

The only factor not definitely specified in setting up this equipment is the equivalent head in feet between grate and face of hot air outlet which should be provided.

This head can only be ascertained by setting up a duplicate furnace in a typical one, two, or three-story piped-furnace plant and determining the total amount of air handled by natural circulation for an average temperature rise from inlet to bonnet of 130 deg. fahr. With this quantity of air known the outlet riser is

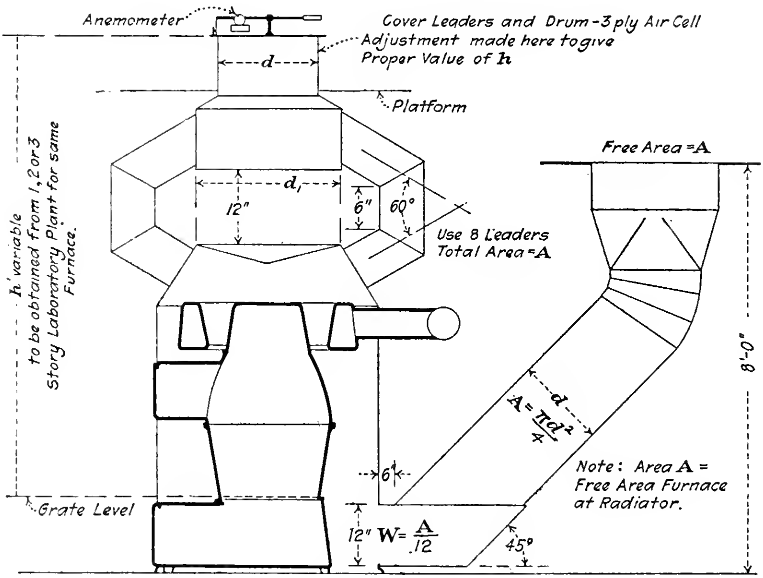

FIG. 3. EQUIVALENT TESTING PLANT FOR PIPED FURNACE.

adjusted to give this same capacity at the same temperature rise. Values for the proper height of the equivalent testing plant (Fig. 3) will depend only on the resistance of the leaders, stacks and registers in one, two and three-story plants. Once these heights are determined for systems in one, two and three-story buildings, the equivalent testing plan will truly represent the actual resistance in such buildings.

The observed data and calculated results are then reported on the Data and

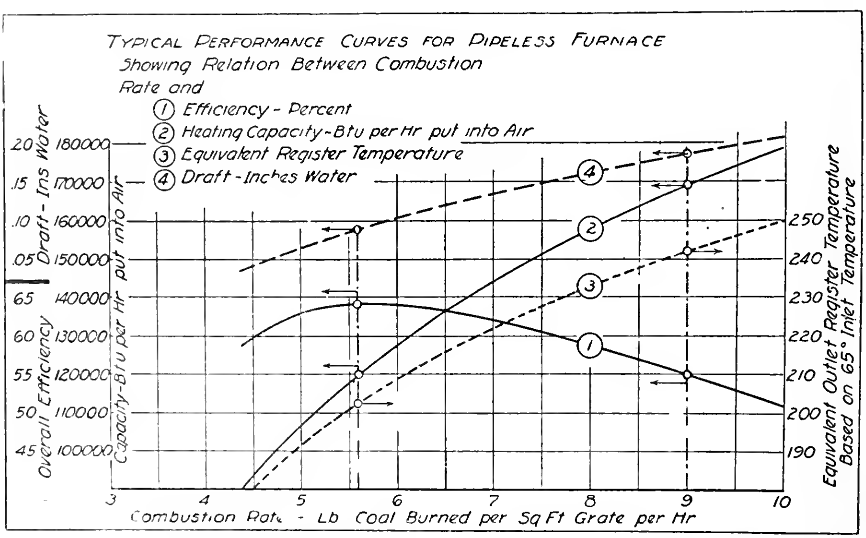

FIG. 4. TYPICAL PERFORMANCE CURVES FOR PIPELESS FURNACE.

Result Sheets for Piped Furnace Test given below. Fig. 3 not only shows the general set-up of the equipment for a piped furnace test, but can also be made to serve as a graphical result sheet similar to Fig. 2 by addition of the necessary items.

Typical performance curves from a series of such tests as are described above for a pipeless furnace are shown in Fig. 4.

DATA AND RESULT SHEET—PIPELESS FURNACE TEST

At.............................. Date................................

GENERAL DATA

Built by....................................Builder's No.................
Type of Furnace...
Type of Inner and Outer Casings...................................
....
Type of Grates..
Type of Chimney...
Type of Register..
Rated Capacity..

DIMENSIONS

Fire pot diameter at top.....................in., at bottom...................in.
Grate area...sq. ft.
Diameter of outer casing...in.
Diameter of inner casing, net inside.............................in.
Thickness of inner casing..in.
Diameter of radiator...in.
Distance from outside of radiator to inner casing................in.
Distance between inner and outer casing..........................in.
Distance from floor to bottom of inner casing....................in.
Least free area across inner casing.................sq. in..................sq. ft.
Least free area between inner and outer casing..........sq. in.............sq. ft.
Ratio of cold air free area to hot air free area.................
Throat diameter below register net...............................in.
Width and length of cold-air register............................in.
Net free area of cold-air register.................sq. in..................sq. ft.
Diameter of hot-air register......................................in.
Net free area of hot-air register..................sq. in..................sq. ft.
Ratio free area to gross area of register...........cold air...........hot air
Ratio free area of cold air to free area of hot air register.....................
Diameter and length of smoke pipe..................in. x..................ft.
Size and height of chimney.....................in. x ....ft.

No.	*Name of Item with Units*	
1.	Duration of test, hours	....
2.	Barometer, in. of mercury	....
3.	Kind of coal	....
4.	Size of coal	....
5.	Approximate analysis of coal as fired, per cent	....
6.	Fixed Carbon	....
7.	Volatile Matter	....
8.	Moisture	....
9.	Ash	....
10.	Sulphur, separately determined	....
11.	Calorific value of coal as fired, by oxygen calorimeter, B.t.u. per lb.	....
12.	Ultimate analysis of coal as fired, per cent	....
13.	Carbon	....
14.	Hydrogen	....
15.	Oxygen	....
16.	Nitrogen	....

17. Sulphur ....
18. Moisture ....
19. Ash ....
20. Analysis of dry refuse at end of test, per cent.......... ....
21. Fixed carbon ....
22. Volatile matter ....
23. Earthy matter ....
24. Calorific value B.t.u. per lb.......... ....
25. Draft at smoke outlet, in. of water.......... ....
26. Temperature of outside air, deg. fahr.......... ....
27. Temperature of air entering ash pit, deg. fahr.......... ....
28. Temperature of inlet air at register face, dry bulb, deg. fahr.... ....
29. Temperature of inlet air at register face, wet bulb, deg. fahr... ....
30. Temperature of outlet air at register face, deg. fahr.......... ....
31. Temperature rise of air from inlet to outlet, deg. fahr.......... ....
32. Equivalent outlet temperature at register face above 65 deg. fahr. ....
33. Temperature of cold air 2 in. above bottom of inner casing, deg. fahr. ....
34. Temperature of outer casing opposite center of fire pot, deg. fahr. ....
35. Temperature of outer casing 6 in. above floor, deg. fahr.......... ....
36. Temperature of flue gas, deg. fahr.......... ....
37. Velocity through free area of outlet register, ft. per min.......... ....
38. Velocity through minimum free area of furnace, ft. per min.... ....
39. Velocity through minimum free area of outer casing, ft. per min. ....
40. Velocity through free area of inlet register, ft. per min.......... ....
41. Volume of air leaving hot-air register, measured at the actual register temperature, cu. ft. per hour.......... ....
42. Volume of air leaving hot-air register, measured at the equivalent register temperature, cu. ft. per hr.......... ....
43. Density of air entering cold air register, lb. per cu. ft.......... ....
44. Density of air leaving hot air register, lb. per cu. ft.......... ....
45. Weight of air circulated per hr., lb.......... ....
46. Weight of air circulated per lb. of coal burned, lb.......... ....
47. Weight of air circulated per 10,000 B.t.u. supplied in coal, lb.. ....
48. Weight of coal fired, total, lb.......... ....
49. Weight of dry refuse at end of test, total, lb.......... ....
50. Weight of equivalent coal in refuse, lb.......... ....
51. Net weight of coal burned during test, lb.......... ....
52. Combustion rate, lb. of coal burned per sq. ft. of grate per hr.. ....
53. Heat developed by net coal burned per hr., B.t.u.......... ....
54. Heat put into air between inlet and outlet per hr., B.t.u.......... ....
55. Heat available above 70 deg. fahr. for heating house per hr., B.t.u. ....
56. Heat put into air between inlet register and bottom of inner casing per hr., B.t.u.......... ....
57. Per cent of total heat put into air which is transmitted by inner casing to entering air.......... ....
58. Overall efficiency of furnace, per cent.......... ....
59. Per cent carbon dioxide in flue gas.......... ....
60. Per cent oxygen in flue gas.......... ....
61. Heat lost in flue gas per lb. of coal burned, B.t.u.......... ....
62. Heat lost by radiation and *unaccounted for* per lb. of coal burned, B.t.u. ....
63. Heat lost in flue gas (approx.), per cent.......... ....
64. Heat lost by radiation and *unaccounted for*, per cent.......... ....

Remarks:

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DATA AND RESULT SHEET—PIPED FURNACE TEST

At.............................. Date..............................

GENERAL DATA

Built by ....Builder's No. ....
Type of Furnace..
Type of Outer Casing..
Type of Lining..
Type of Grates..
Type of Chimney...
Number of Leaders...
Rated Capacity ....

DIMENSIONS

Fire pot diameter at top...................in., at bottom....................in.
Grate area ....sq .ft.
Diameter of outer casing...in.
Diameter of lining, net outside...in.
Thickness of lining ....in.
Diameter radiator ....in.
Distance from outside of radiator to lining...................................in.
Diameter cold air duct..in.
Height cold air shoe......................in., and width.....................in.
Least free area across inner casing.................sq. in. ....sq. ft.
Ratio of cold air free area to total hot air leader area..........................
Throat diameter of outlet...in.
Width and length of cold-air register.................in. x ....in.
Net free area of cold-air register.................sq. in. ....sq. ft.
Ratio free area to gross area of cold air register...............................
Diameter of top of bonnet..in.
Diameter of leaders..in.
Total area of leaders......................sq. in. ....sq. ft.
Diameter of bottom of drum...in.
Height from grate to face of hot-air outlet...............ft. ....in.
Diameter and length of smoke pipe....................in. ....ft.
Size and height of chimney.......................in. ....ft.

No.	*Name of Item with Units*	
1.	Duration of test, hours..............................	....
2.	Barometer, inches of mercury....................	....
3.	Kind of coal..	....
4.	Size of coal...	....
5.	Approximate analysis of coal as fired, per cent...............	....
6.	Fixed Carbon	....
7.	Volatile Matter	....
8.	Moisture	....
9.	Ash	....
10.	Sulphur, separately determined	....
11.	Calorific value of coal as fired, by oxygen calorimeter, B.t.u. per lb.	....
12.	Ultimate analysis of coal as fired, per cent....................	....
13.	Carbon	
14.	Hydrogen	....
15.	Oxygen	....
16.	Nitrogen	....
17.	Sulphur	....
18.	Moisture	....
19.	Ash	....
20.	Analysis of dry refuse at end of test, per cent...............	....
21.	Fixed carbon	....
22.	Volatile matter	....
23.	Earthy matter	....

24. Calorific value B.t.u. per lb.............................. ....
25. Draft at smoke outlet, in. of water........................... ....
26. Temperature of outside air, deg. fahr......................... ....
27. Temperature of air entering ash pit, deg. fahr................ ....
28. Temperature of inlet air at register face, dry bulb, deg. fahr.... ....
29. Temperature of inlet air at register face, wet bulb, deg. fahr.... ....
30. Temperature of outlet air at face hot-air outlet, deg. fahr...... ....
31. Temperature of air at bonnet, deg. fahr........................ ....
32. Temperature rise of air from inlet to bonnet, deg. fahr........ ....
33. Equivalent outlet temperature at register faces above 65 deg. inlet, deg. fahr. allowing .... deg. fahr. drop in leaders and stacks ....
34. Temperature of outer casing opposite center of fire pot, deg. fahr. ....
35. Temperature of outer casing 6 in. above floor, deg. fahr...... ....
36. Temperature of flue gas, deg. fahr............................ ....
37. Velocity through free area of outlet, ft. per min............. ....
38. Velocity through minimum free area of furnace, ft. per min.... ....
39. Velocity through cold air duct, ft. per min................... ....
40. Velocity through free area of inlet register, ft. per min...... ....
41. Volume of air leaving hot-air outlet, measured at the actual outlet temperature, cu. ft. per hr..................... ....
42. Volume of air leaving hot-air registers, measured at the equivalent register temperature, cu. ft. per hr................ ....
    (Note:—See item 33.)
43. Density of air entering cold air register, lb. per cu. ft.......... ....
44. Density of air leaving hot air outlet, lb. per cu. ft............. ....
45. Weight of air circulated per hr., lb........................... ....
46. Weight of air circulated per lb. of coal burned, lb............ ....
47. Weight of air circulated per 10,000 B.t.u. supplied in coal, lb.... ....
48. Weight of coal fired, total, lb................................ ....
49. Weight of dry refuse at end of test, total, lb................. ....
50. Weight of equivalent coal in refuse, lb....................... ....
51. Net weight of coal burned during test, lb..................... ....
52. Combustion rate, lb. of coal burned per sq. ft. of grate per hr. ....
53. Heat developed by net coal burned per lb., B.t.u................ ....
54. Heat put into air between inlet and bonnet per hr., B.t.u......... ....
55. Heat available above 70 deg. fahr. for heating house per hr., B.t.u. (Note:—See item 33.) ....
56. Overall efficiency of furnace, per cent (Note:—Based on bonnet temperature.) ....
57. Per cent carbon dioxide in flue gas.......................... ....
58. Per cent oxygen in flue gas.................................. ....
59. Heat lost in flue gas per lb. of coal burned, B.t.u............. ....
60. Heat lost by radiation and *unaccounted for* per lb. of coal burned, B.t.u. ....
61. Heat lost in flue gas (approx.) per cent...................... ....
62. Heat lost by radiation and *unaccounted for*, per cent.......... ....

Remarks:

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No. 584

RADIAL BRICK CHIMNEYS

By W. F. Leggo[1], New York, N. Y.

Non-Member

THE wonder to us all is how man ever invented or built what has now developed to be the modern chimney. There is a legend that tells us that the old prime minister of feasts was directed by the king in the old Grecian days to prepare for a great festival. The old prime minister, known as Ab, hastened to get ready. It had always been the custom to build the fires for these banquets in great pits in the ground, but Ab could not use them, for it had rained for many days and the pits were filled with water. Ab was worried—his hut was drenched and watersoaked, but as the banquet must be made ready, there was only one thing to do and that was to build a fire in the center of his hut. This was done, and soon the blazing fire drove Ab and his slaves out of the hut. The game was all skinned ready for the feast, so they waited outside the hut for the smoke to die out, so they might rush in and throw on the meat. But, as Ab stood watching his home belch smoke, he suddenly noticed a little cloud of smoke curl up from the center of the roof, and then more and more smoke followed. Ab surely expected the walls of his house to collapse, but instead, the superheated air rose in a thin column toward the roof, and, concentrated in one spot, had burned a hole in the roof, and the smoke rushed through. Nature had helped man build his first chimney.

GREEKS CREDITED WITH FIRST CHIMNEY

The first chimney (Fig. 1) was introduced in 300 B.C. and used by the Greeks in a crude form of smelter. In the ruins of Pompeii (Fig. 2) there was found a chimney in a bakery—75 A.D. Venice was one of the first cities to adopt the chimney for in 1347 many of them were overthrown in the earthquake.

In the fifteenth and sixteenth centuries, we find that many chimneys (Fig. 3) were built of stone. We also had the elaborate type of chimneys on the castles, etc., in England during the sixteenth century. We find that most of these chimneys were very handsome, carrying out the general architectural design of the buildings with rich carvings, matching the doorways and windows. It seems evident, however, that very few of these chimneys were ever more than ornaments, for it is recorded that

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Paper originally presented before New York Chapter; also at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1921.

people still continued to carry the hot brick to church in order to keep their feet warm.

The chimney has played a wonderful part in the development of mankind. We have the great Santa Claus, who could never have lived without the chimney and even in our modern apartment today, we still have to get old St. Nick down the chimney. We had the London chimney sweeps, and many a writer has used the open fireplace and chimney as the symbol of hospitality. Smoke coming from the house-top chimney always denotes welcome.

RULE OF THUMB GOVERNED CHIMNEY DESIGN

The commercial chimney came into use about the time of the power engine, about 1706 (Fig. 4). For over 150 years the only material used was brick of the old type (what we now call common brick) or in some cases stone was used. The design of all these chimneys was more or less by rule of thumb method. Having guessed how high the chimney was to be, which generally was just enough to carry it above the roofs

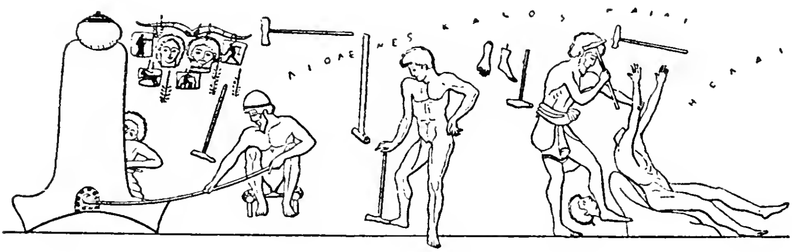


FIG. 1. GREEK CHIMNEY, 300 B.C.

of adjoining buildings, the diameter was assumed on a fairly accurate basis. The design of the old common-brick chimney was about like this: The same inside dimension at both top and bottom of chimney, and the outside dimensions figured on the basis of having the wall 12 in. thick at the top and increasing 4 in. or the thickness of one brick, every 15 to 20 ft. It goes without saying that these chimneys must stand up, in fact many of them are still performing service today. However this was an expensive method, and when the radial brick was introduced, it was a big task to educate people to use the new type of brick. The majority of these old common-brick chimneys was square in shape with square flues, but modern chimneys of today are built of course with circular flues, which offer less resistance to the flow of the gases.

Today we have the most scientific type of brick for the chimney (Fig. 5). The bricks are moulded to fit a circle and the standard face dimensions of chimney brick are 6½ in. wide and 4⅝ in. high. Having all brick of these dimensions on the face, makes the outside appearance of the brickwork of the chimney uniform. The bricks are made in five standard lengths, ranging from approximately 4¼ in. up to 10¾ in. All the bricks are perforated, the holes being approximately 1 in. by 1½ in. and in the small brick there are four holes and in the largest brick

10 holes. The design of these perforations is figured on the basis of having the area of them not to exceed 25 per cent of the cross area of the brick. In order to secure additional surface, the sides of the brick are corrugated, the corrugations being about ¼ in. wide and ⅛ in. deep. This insures a stronger mortar joint in the brickwork. These bricks are so bonded as to make up the several wall thicknesses, changing the bond in the brickwork every three courses as shown in Fig. 6. In the bonding of the brickwork, three courses of one size brick are laid on the outside

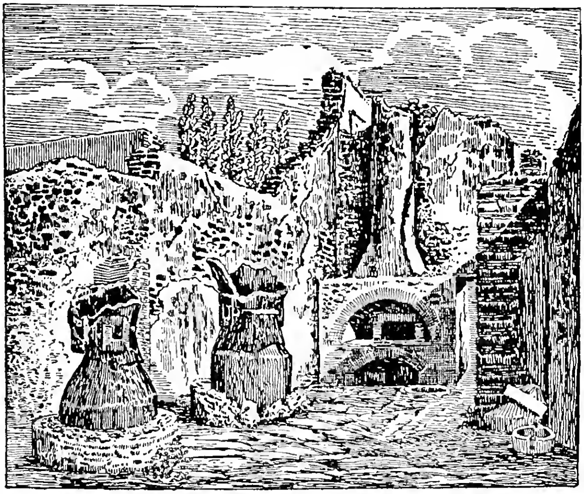


FIG. 2. CHIMNEY IN A BAKERY AT POMPEII, 75 A.D.

and backed up with one or two of the smaller size brick on the inside as shown in this picture.

EACH CHIMNEY A SEPARATE PROBLEM

Some engineers seem to think that the subject has been so standardized that a chimney company ought to be able to give on immediate notice all the details of design, etc. This is not possible for every plant requires a chimney to perform a definite service and all these items have to be considered. In determining the size stack to use, the engineer must be sure to build the chimney sufficiently large to take care of any additional units to the boiler plant and at the same time, select the most economical size for the operation of the boilers. The first thing therefore to consider is the draft required. For general practice the assumptions for draft losses are about as follows:

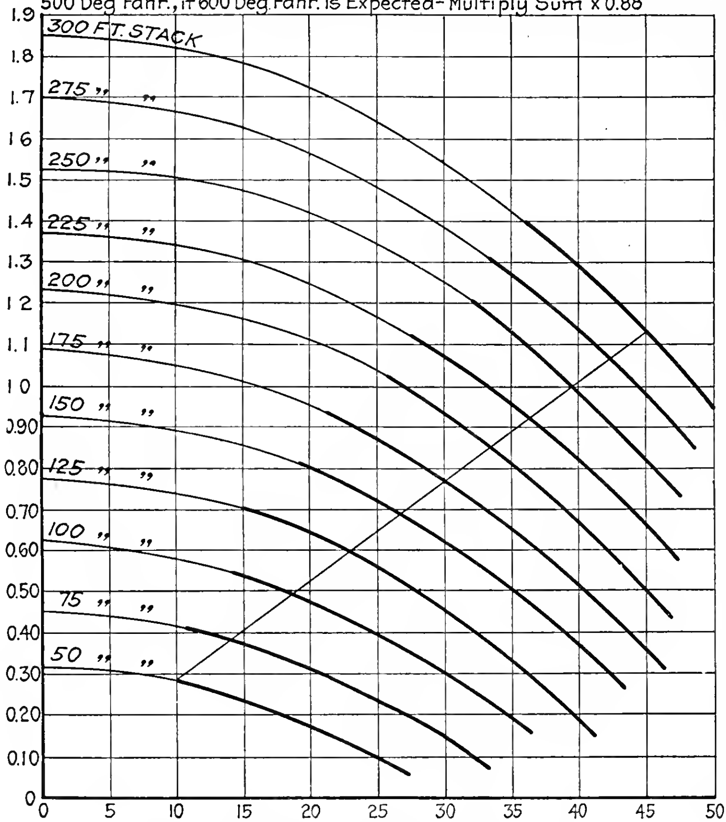
Loss in boiler..........0.3 in.
Loss in furnace........0.5 in.
Flues ....0.05 in. for 100 ft. of flue plus right-angle bends figured at 0.02 in.

For example, let us assume a 150 ft. x 6 ft. chimney which is the standard size generally used for 1000 hp. of boilers. Assuming that the plant has two 500 hp. boilers, we can figure that the draft loss through the boiler

TABLE NO.1 - DRAFT REQUIRED AT BASE OF STACK

or

Sum of Resistances of Boiler Flue and Fire Based on Flue Temperature of 500 Deg Fahr., if 600 Deg. Fahr. is Expected - Multiply Sum x 0.88



Velocity of Gas in Stack, Ft. per Sec

Lb. of Coal to be Burned per Hour

STACK DIAM.	10	15	20	25	30	35	40	45
4	835	1250	1670	2090	2510	2930	3350	3760
4.5	1080	1620	2060	2700	3240	3780	4320	4850
5	1360	2040	2720	3400	4080	4750	5440	6100
5.5	1660	2500	3330	4160	5000	5820	6660	7500
6	2010	3010	4010	5020	6020	7030	8040	9000
6.5	2380	3560	4750	5950	7140	8310	9500	10700
7	2780	4180	5570	6960	8350	9750	11140	12500
7.5	3220	4820	6440	8040	9660	11250	12870	14500
8	3680	5540	7360	9220	11030	12880	14720	16500
8.5	4180	6280	8360	10450	12530	14620	16710	18800
9	4700	7050	9400	11750	14100	16500	18810	21200
10	5860	8780	11700	14650	17600	20500	23400	26300
11	7140	10700	14300	17850	21400	25000	28600	32100
12	8530	12800	17050	21300	25600	29800	34100	38400

Based on 500 Deg. Stack Temp. - for 600 Deg., Lb. of Coal x 1.10

and furnace would be 0.8 in. For the two right-angle bends in the flues and approximately 50 ft. of steel breeching connection, the draft loss through the flue would be approximately 0.07 in. This gives us a total draft loss of approximately 0.87 in. and it will be noted in Table 1 that a 150 ft. stack gives a draft of 0.9 in. at the flue opening; therefore it is safe to assume that 150 ft. is the correct height.

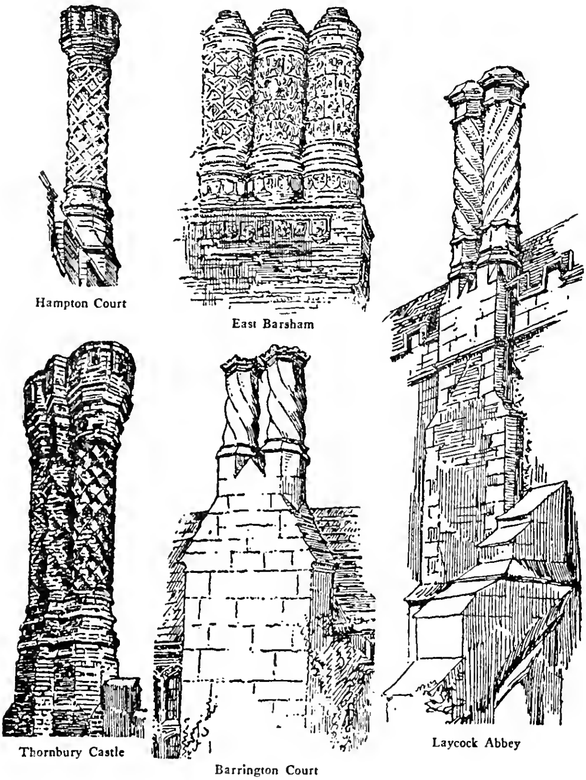


FIG. 3. TUDOR BRICK CHIMNEYS, ENGLAND, SIXTEENTH CENTURY

We also have for the convenience of the engineering world a table of horsepower giving sizes of chimney required (Table 2). In using this table, it is always well to adhere to the middle of the table as, for instance, for 250 hp., the correct size would be 80 ft. by 3 ft. 6 in.

Every chimney design must be figured out accurately for each particular installation. The radial-brick chimney built today is guaranteed to withstand a 100 mile wind and the lining should be designed for the proper temperature.

Having determined the size of chimney, its design is the next thing to be considered. Fig. 7 shows what is known as the standard type of

TABLE 2. SIZES OF CHIMNEYS REQUIRED FOR VARIOUS HORSE-POWER CAPACITIES

Diam in inches	Area "A" Sq. Ft.	HEIGHT OF CHIMNEY 50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	225 ft.	250 ft.	275 ft.	300 ft.
		Horse Power of Chimney Based on Christie's Formula.														
24	3.14	72	79	85	91	97										
27	3.98	91	100	108	116	123										
30	4.91	113	123	133	143	151	159									
33	5.94		149	161	172	183	193	202								
36	7.07		178	192	205	218	230	241	256							
39	8.30			225	241	256	270	283	302							
42	9.60			261	279	296	312	328	349	383						
48	12.57				365	388	408	429	456	500						
54	15.90					490	517	542	577	633	684					
60	19.64					605	638	670	713	783	845					
66	23.76						772	811	863	946	1021	1091				
72	28.27						919	965	1026	1125	1217	1300	1378			
78	33.18							1132	1204	1320	1427	1525	1618	1704		
84	38.48							1311	1398	1530	1651	1768	1874	1976	2070	2165
90	44.18								1604	1758	1899	2030	2153	2270	2380	2485
96	50.27								1828	2000	2163	2213	2451	2583	2710	2832
102	56.75								2060	2258	2438	2608	2764	2913	3052	3191
108	63.62								2310	2530	2735	2920	3100	3267	3424	3580
114	70.88									2820	3048	3260	3455	3640	3820	3990
120	78.54									3128	3380	3612	3830	4035	4230	4425
132	95.03									3780	4084	4370	4672	4882	5120	5350
144	113.10									4505	4865	5200	5320	5818	6095	6375
156	132.73										5700	6100	6470	6816	7150	7468
168	153.94										6615	7075	7500	7910	8380	8660
180	176.71										7600	8127	8615	9080	9520	9950
192	201.06										8650	9250	9800	10320	10830	11310
204	226.98										9760	10430	11060	11660	12220	12780
216	254.47										10930	11735	11990	12640	13250	13850
228	283.53										12170	13000	13800	14530	15230	15930
240	314.16										13500	14450	15320	16130	16910	17680

For pounds of coal burned per hour for any given size of chimney, multiply the figures in the table by 4.

The formula for this table is horse-power $= 3.25\ A\ V\ H$

based on 4 lb. of coal burned per horse-power per hour. For the Middle States and Western bituminous coal, it is recommended that the height as found by this table be maintained, but the area be increased 25 per cent. For example: What size chimney for 2000 hp. Eastern coal? Adhering to center of table, the size is found to be 175 ft. x 8 ft., to be on the safe side. For Western coal, it would be 175 ft. x 8 ft. plus 25 per cent, or 175 ft. x 9 ft. It must be observed, however, that temperatures, flues, type of boilers, economizers and other accessories may have a very decided influence on the proper size.

radial-brick chimney round construction for the entire height, that is, that radial brick is used throughout the entire structure. This is the correct size of chimney to use in a plant of 1500 boiler hp., using western coal. The plant consists of three 500 hp. boilers used in connection with economizers. According to Table 2 showing horsepowers of boilers and heights of chimney, the size of chimney required would be 150 ft. by 6 ft. 9 in. and on account of using western coal, the area should be increased 25 per cent, making it 7 ft. 6 in. and the height increased from 150 ft. to 180 ft. in order to take care of the economizers, etc.

In designing radial-brick chimneys, the standard practice is to allow compression in the brickwork up to 35,000 lb. per sq. ft. This is not excessive for the reason that all hard-burned chimney brick has a crushing strength of about 6000 lb. per sq. in.

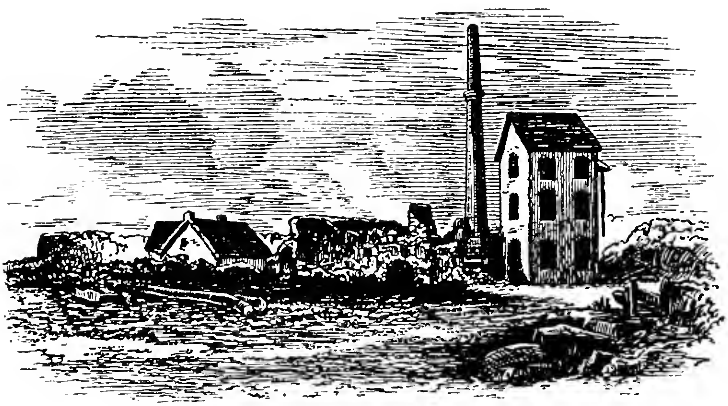


FIG. 4. STACK USED WITH SAVERY'S ENGINE, 1705

Practically all the radial-brick chimneys designed are of the tension type. It is most expensive to build a no-tension, or gravity chimney, and to our minds this is a great waste of good materials. The allowable tension should not exceed 5000 lb. per sq. ft. in chimneys 150 ft. and higher, but in smaller chimneys, it could run as high as 6000 lb. per sq. ft. These limits are all figured on the basis of wind with velocity of 100 miles per hour, figuring the pressure on the brickwork at 25 lb. per sq. ft. on the projected area. Radial brick chimneys have been designed according to these lines for over 30 years without a failure.

Where it is necessary to have an extra-wide flue opening, it is well to build the lower portion of the chimney with a pedestal having a square or octagonal base as shown in Fig. 8. For general practice, it is not safe to have the flue opening in the round column any wider than one-third of the diameter where the opening enters the chimney but in a square base, it is, of course, safe to make the width of the opening one-half the cross dimension of the base and with octagonal bases it is safe to cut out the entire flat of the octagon.

CHIMNEY LININGS

For ordinary boiler purposes, where the average temperature does not exceed 600 to 800 deg., the lining in the chimney should be approximately one-fifth of the height, this lining to be built resting on a corbel racked out from the shell of the stack about 2 ft. below the flue opening and to be built with radial brick. As all radial brick is burned at a temperature of about 2000 to 2300 deg., radial brick is very satisfactory for the ordinary boiler-house chimney. There are however many special cases where the temperature of the gases will run exceedingly high, such as in smelters, steel furnaces, etc. There is only one type of lining to use for a chimney where the temperature ranges about 2000 deg. to 2500 deg. There are now being built 10 chimneys for the Government armor-plate plant, designed to take care of the gases from open-hearth, reheating and

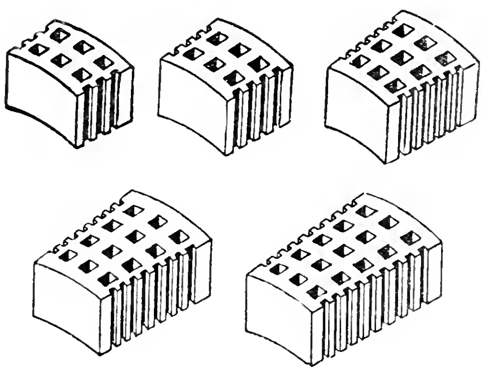


FIG. 5. TYPES OF MOULDED BRICKS FOR CONSTRUCTION OF RADIAL BRICK CHIMNEYS

plate-mill furnaces. The normal temperatures of these gases will be approximately 1200 deg. except when they burn out the furnaces and at that time temperature jumps from 1200 to well over 2000 deg. in a very short period. The design of the lining is what is known as an independent fire brick; starting with the bottom of the flue opening that goes in the foundation, the lining is built up of fire brick in independent sections. No one section of the lining exceeds 40 ft. in height. Unlike the linings in the ordinary boilerhouse chimneys, which are laid up in cement-lime mortar as in the outside shell, the linings for high-temperature chimneys are laid up in air-drying high-temperature cement. It is very necessary to use an air-drying cement, as it would be impossible to build the linings using a high-temperature cement where it would be necessary to secure a high temperature before the cement will flux and make a tight joint. For many years this matter of high-temperature chimneys has been discussed and it is only within the last eight years that owners have been willing to pay the high cost in order to build the chimney with an independent lining as shown.

In passing it might be well to consider the foundation required for the normal chimney, which is specified in Table 3.

The best radial brick is made in the Clearfield District of Pennsylvania and also in the middle section of Ohio. Wherever coal is found, the shale rock underneath it will burn to a beautiful buff color. In Ohio,

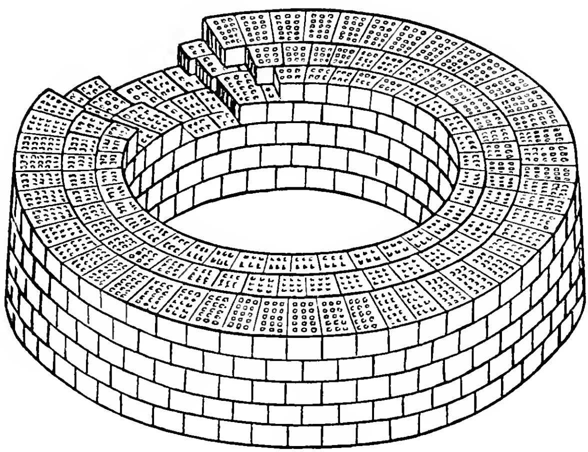


FIG. 6. METHOD OF BONDING MOULDED BRICKS IN CONSTRUCTING RADIAL BRICK CHIMNEYS

the majority of brick manufactured is red in color due simply to the fact that in order to secure the coal, it is necessary to go very deep, and after a plant has been established a long time and the top shale rock has all been removed, it makes a very simple matter to mine the coal

TABLE 3. NORMAL TOTAL DEPTH OF FOUNDATION BASED ON BEARING VALUE OF SOIL AT 2 TONS PER SQ. FT.

SIZE OF CHIMNEY			Total depth of foundation
75′ x 3′	up to and including	100′ x 6′	4′–6″
100′ x 7′	" " " "	100′ x 8′	5′–0″
125′ x 3′	" " " "	125′ x 8′	5′–0″
125′ x 8′ 6″	" " " "	125′ x 10′	6′–0″
150′ x 4′	" " " "	150′ x 8′	6′–0″
150′ x 8′ 6″	" " " "	150′ x 10′	6′–6″
175′ x 4′	" " " "	175′ x 9′	6′–9″
175′ x 10′	" " " "		7′–0″
200′ x 7′	" " " "	200′ x 9′	8′–0″
200′ x 10′	" " " "		9′–0″
225′ x 9′	" " " "	225′ x 10′	10′–0″

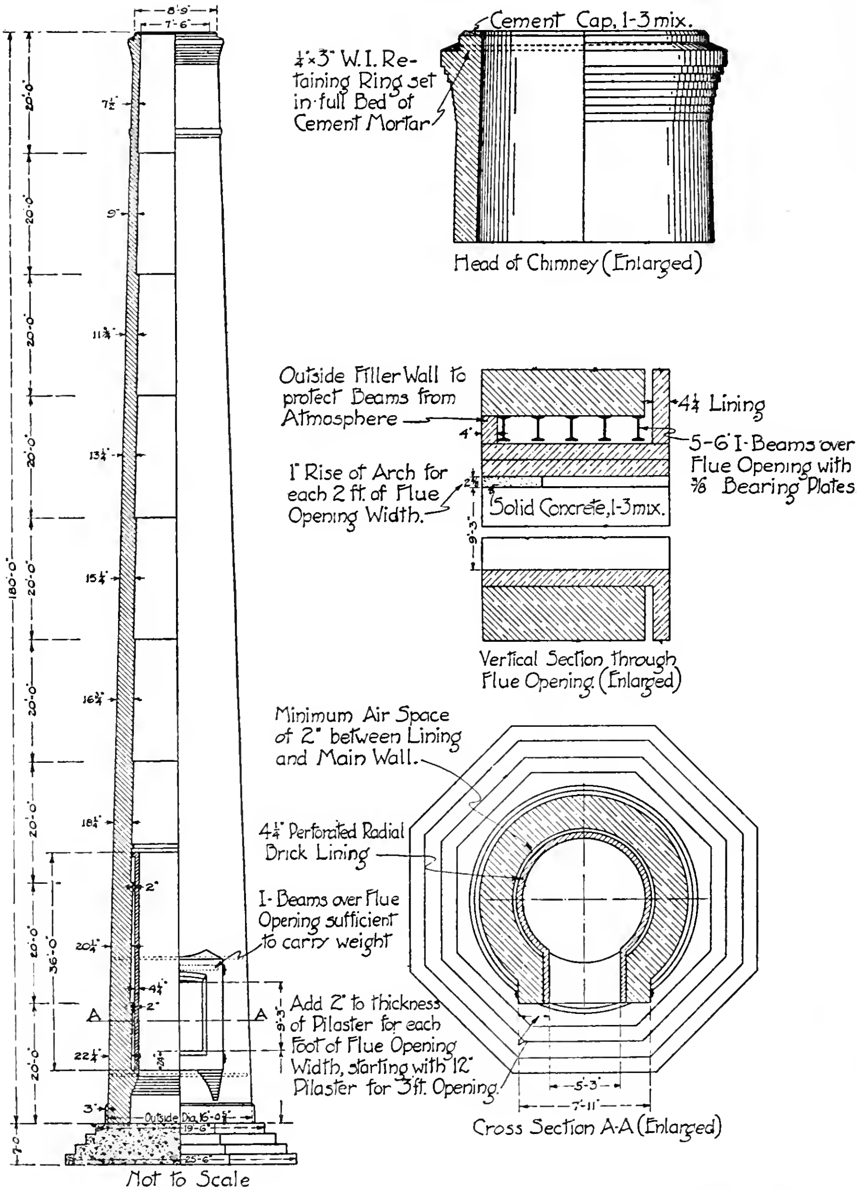


FIG. 7. DETAILS OF A TYPICAL RADIAL BRICK CHIMNEY, ROUND FOR ITS ENTIRE HEIGHT

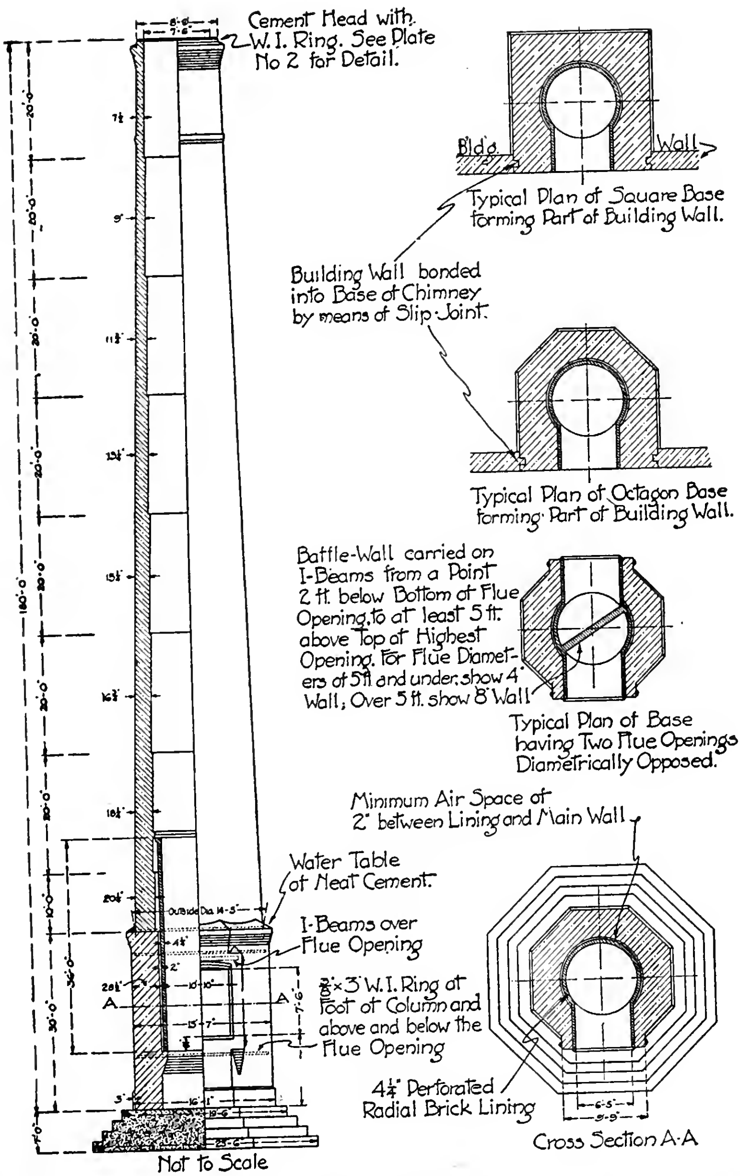


FIG. 8. DETAILS OF RADIAL BRICK CHIMNEY WITH ROUND COLUMN ON OCTAGONAL BASE

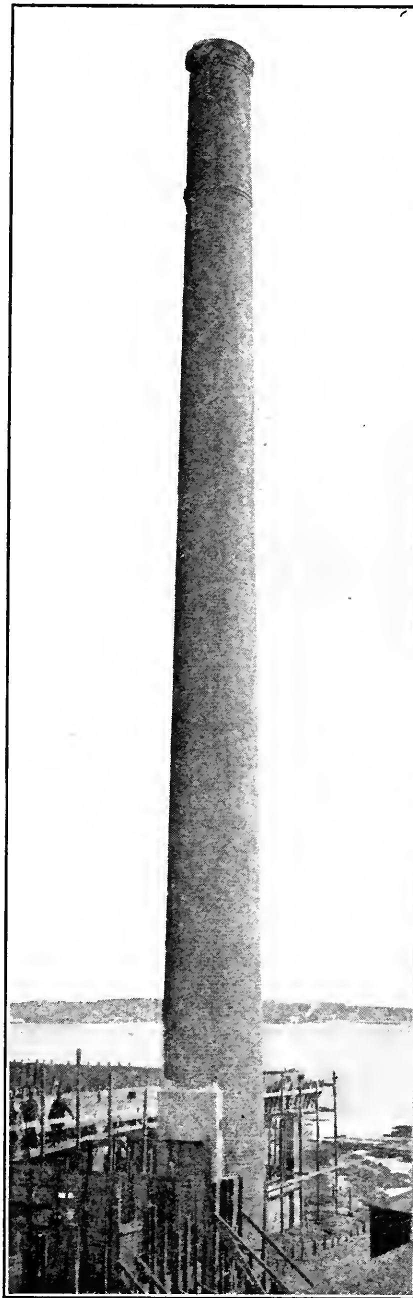


FIG. 9. RADIAL BRICK CHIMNEY AT THE PAIRPOINT CORPORATION, NEW BEDFORD, MASS., 125 FT. BY 4½ FT.

FIG. 10. RADIAL BRICK CHIMNEY AT TOTTENVILLE COPPER CO., TOTTENVILLE, N. Y., 200 FT. BY 7 FT.

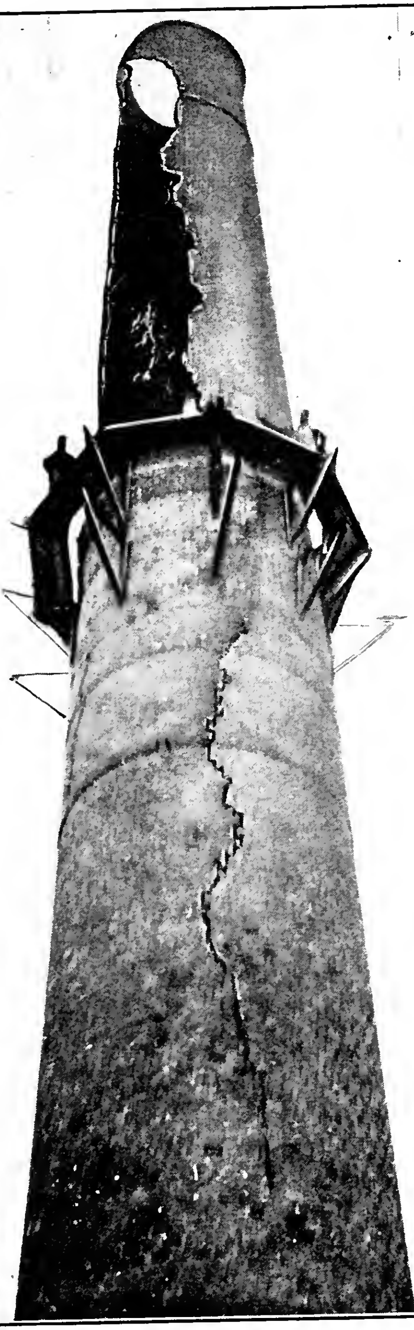


FIG. 11. RADIAL BRICK CHIMNEY DAMAGED BY INTERNAL EXPLOSION—REPAIRED WHILE IN OPERATION

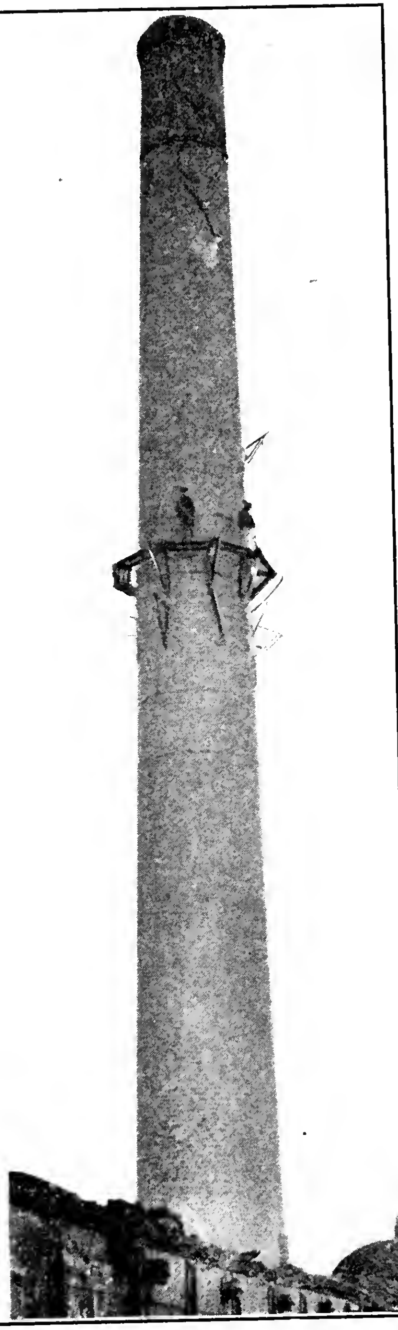


FIG. 12. VIEW OF REPAIRS TO DAMAGED CHIMNEY ALMOST COMPLETED

and secure shale and clay that will burn buff in color. It will therefore be noted that there is practically no difference in the quality of the material whether it be red or buff, and it is just a matter of taste. Some people still ask for grey and speckled radial brick but these are only chance products and are very hard to obtain.

Radial brick chimneys are used on boiler installations, incinerators, furnaces, school and college buildings, public institutions, etc. With the use of terra cotta or stone trimmings, it is possible to build a very elaborate radial-brick chimney to coincide with the architecture of any buildings.

No. 585

STEEL SMOKE STACKS

By W. E. Goldsworthy[1], New York, N. Y.

Non-Member

THERE are several matters in regard to the construction of steel stacks as commonly designed and built that I believe can be changed or modified to great advantage. To most of us a stack is a simple thing, a steel pipe stood on end. What could be simpler? What is there to know about such a simple affair?

In reality, however, there is much to know, while to properly specify or design an adequate steel stack calls for unusual experience or special information.

FOUR PRINCIPAL TYPES

From the viewpoint of a stack manufacturer, there are four principal types of smoke stacks: 1. The self-supporting stack. 2. The guyed stack. 3. The interior office-building stack. 4. The exterior office-building stack.

The *self-supporting stack* is the largest and most difficult stack to design. It is often built of great diameter and height—sometimes 20 to 25 ft. in diameter and 200 to 300 ft. high. Stacks of this type are attached by anchor bolts to a concrete base or sometimes in power houses to a special steel support. To attempt to go into the details of the design of this type of stacks would be outside of the purposes of this paper but a few words about some of the most prominent features of these stacks may be of interest and value.

The stresses in a self-supporting stack are largely dependent on the ratio between the diameter and the height. If the ratio does not exceed 1 to 20, the stresses will not be high for a stack of ordinary proportions; for instance in a stack 5 ft. in diameter by 100 ft. high, or 6 ft. diameter by 125 ft. high, the unit stresses will be low and an economical stack can be built. This does not hold so nearly true on the large diameters, but is correct up to 10 ft. diameter. It is usual to flare the bases of these stacks, the proportions of the flare being usually as follows: Commencing at the point about one-sixth of the height from the bottom the diameter is increased until at the bottom it is one and one-half times the diameter.

It was formerly the custom to use heavy cast-iron bases to transmit the stresses from the stack into the foundation and rivet the stack directly to the base. This is not good practice however as this puts

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Paper originally presented before New York Chapter; also at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1921.

some of the base in tension which is poor engineering. The better way is to rivet special steel lugs, made of angles, onto the stack, thus transmitting the strain through shear in the rivets. The cast-iron base can be safely omitted, substituting an angle ring about the bottom instead. Designed in this manner the stack costs less and is stronger.

The proper design of this type of stack is a highly technical affair, and normally it is better to specify the diameter and height and let the expert work out the details. No steel less than ¼ in. in thickness should be used as there is no economy in using the lighter metals, since fabrication and erection cost about the same for either. Fancy tops or astragals are expensive and usually wear out before the remainder of the stack.

OPPORTUNITY FOR IMPROVEMENT

Guyed stacks are a much simpler proposition, but it is in the design of these that there is much opportunity for improvement. The self-supporting stack, owing to its design, calls for heavier plates in the lower part than in the upper part—this undoubtedly led originally to the adoption of the same custom in guyed stacks. It has been as long as one can remember, and still is, the custom to build guyed stacks of different thicknesses throughout, the upper parts lighter than the lower. It is common to read of a guyed stack, say 5 ft. in diameter by 100 ft. high, made of ⅜ in. plate for 25 ft., of $\frac{5}{16}$ in. for 25 ft., of ¼ in. for 25 ft. and of $\frac{3}{16}$ in. for 25 ft. The engineer specifying this would probably lean back in his chair after completing the specification and feel that he had called for a good job. I want to say that if he turned the stack upside down, he would then have a stack which would last two to three times as long and be amply strong. The great majority of stacks of this type which have been built, are built wrong and have wasted the money of the men who paid for them.

The reason for this is that the upper part of a stack wears out first. There is a point, usually within the upper 30 ft. of a stack, where the gases condense and precipitate a mild sulphuric acid on the stack which sets up a rapid corrosion. The author has seen dozens of stacks, both large and small, where this has occurred and there is no room for doubt. The corrosion is usually confined to the upper 25 or 30 ft. of the stack, as the hot gases prevent the precipitated moisture from trickling further down.

Of course, it would seem strange to actually turn these stacks upside down and in practice we do not do this; we build them the same thickness throughout. We usually decide on a thickness of $\frac{5}{16}$ in. or ¼ in., and use that from the bottom to the top. The stresses in guyed stacks are negligible and are only those due to the weight of the steel, as the guys take all of the wind load. In an average stack of this character, the unit stress is from 300 to 500 lb. per in., while it is well known that the steel could safely carry 10,000 to 15,000 lb. per sq. in.

Another point, before leaving this type of stack, is that most engineers specify these stacks to have the lower course telescope into the upper one. It is our practice to reverse this, making the upper course telescope into the lower one. The reason for this is that the stacks built in the old way streak in the first rainstorm, while those built in our way are free from streaking. We accomplish by this simple change, what would cost a lot of money to try and produce in the old way; it would mean doubling up on the riveting and then calking all the seams to get the same result.

It might be in order to mention at this point, that the cost of stack work varies with the number of rivets and the amount of labor called for—hence the fewer the rivets the cheaper the job. Another reason for changing the construction is that the old way provided a little shelf at each joint, for the moisture of condensation to settle on and aid corrosion, while with the changed construction this condition does not exist and corrosion will be retarded.

OFFICE BUILDING STACKS

Interior office building stacks present many problems to the designer or the specification writer. To the best of our knowledge the corrosion in this type of stack is slight. This is probably due largely to the fact that the stack is well insulated and is usually warm for its entire length, doing away with the condensation of the gases of combustion, and is thus protected from both interior and exterior corrosion.

The questions of thickness and style of support are the main questions in this type of stack. For thickness a uniform thickness throughout of $\frac{5}{16}$ in. or $\frac{1}{4}$ in. is good practice. The stacks in the Woolworth Building and in the Equitable Building are $\frac{5}{16}$ in. thick throughout. The Equitable Building stack is 11 ft. in diameter and that in the Woolworth Building, $6\frac{1}{2}$ ft. in diameter.

The best style of support is the method of supporting the stack from the structural steel of the building at about every second or third floor. This involves making the sections 25 to 35 ft. long, or about the length of a gondola car. It is convenient to fabricate, ship, truck and hoist sections this length. Each section when made in this manner takes care of its own expansion if a slip joint is used. It is important to remember that expansion in a stack of this character is a necessary consideration. There could easily be an expansion of 6 to 12 in. in a stack for a high office building.

If the weight of the stack is carried to the ground as is sometimes done and angle joints used, care must be taken to provide flexible ties for maintaining the stack in position and further not to attach the hood solidly to both the stack and the top of the masonry of the stack enclosure.

The exterior office building stack has special conditions of its own. Owing to its length and exposure the gases condense freely and much sulphur-laden moisture is precipitated in them, so that corrosion is quite rapid. It is good judgment to make these quite thick as the cost of shop labor, overhead and erection would not be much greater on the heavier stack than on the light one, while the longevity would be decidedly greater. Stacks of this type should be $\frac{1}{4}$ in., $\frac{5}{16}$ in., or $\frac{3}{8}$ in. thick, depending of course somewhat on the size and should carry their weight to the foundation. Angle joints should be used and the angles should be $\frac{1}{16}$ in. or $\frac{1}{8}$ in. thicker than the material in the stack, as they corrode quite rapidly. These stacks are supported by U-bands at about every story height; these are made loose so that while they support the stack they do not prevent it from expanding. If they were made to fit tight, the stack would undoubtedly tear them off the building and thus bring about its own destruction.

On this type of stack, the upper courses should telescope into the lower ones, otherwise the stacks will streak beautifully and a nice new shiny job will look like second-hand work. Also the stack will last longer if built this way. To those who wonder why a stack cannot be made tight the other way, I will say that it is not practicable to do so; that at the best, the stack would streak somewhat, and that spending a lot of money in extra riveting and calking, will only get a job about 90 per cent tight. Whereas, the method we employ gives a job practically tight without a cent of extra cost.

These exterior stacks usually run above the roof anywhere from 15 to 30 ft. We are often asked to provide guys for these extensions, the designer seeming to fear for their stability. When the stack is adequately braced to the building walls which includes making the top brace especially strong, no guys are necessary and the stresses set up by the wind on the extending part are not great and will be easily carried by any well-designed stack. I might mention a stack which we built for the Robert Gair Co. building in Brooklyn, near the Brooklyn Bridge, which is 10 ft. in diameter and runs 100 ft. above the building without a guy or attachment of any sort.

Exterior stacks should be kept well painted. They should, if possible, be painted every year, preferably with a good quality graphite paint, as this type of paint stands the heat better than most lead paints.

IMPORTANT DETAILS

A few remarks on the size of rivets, rivet spacing, calking and other stack details may be in order. On plates $\frac{3}{8}$ in. thick and thicker, $\frac{3}{4}$ in. rivets should be used; of course, if very thick even larger rivets. For plates $\frac{5}{16}$ in. and $\frac{1}{4}$ in. thick, use $\frac{5}{8}$ in. rivets; $\frac{3}{16}$ in. thick, $\frac{1}{2}$ in. rivets, and No. 10 gauge, $\frac{3}{8}$ in. rivets. The spacing usually adopted is from 3 to 4 in.; the average stack has rivets spaced $3\frac{1}{2}$ in. center to center. Calking is not generally used. The impression is that it adds to the cost without compensating advantages.

Self-supporting and guyed stacks should be provided with means of reaching the top for attaching painter's rigging. Self-supporting stacks are usually provided with a ladder and a painter's trolley; guyed stacks usually with a painter's pulley and a flexible wire reaching to the ground.

All stacks should have a cleanout door at the bottom. Hoods on top of the stacks are unnecessary unless the stack is out of use for long periods. They interfere with the draft and should be avoided when possible.

Stacks for power station and industrial plants are usually round; sometimes, however, in office, loft and apartment buildings, conditions make it necessary to build the stacks oval or rectangular. Measured by area of a cross-section, round stacks are cheapest, then oval stacks and last, rectangular ones. If a rectangular stack is unavoidable make it with rounded corners preferably to about a 3 in. radius. To build rectangular stacks with angle corners is a fine way to waste money. When built in this manner there are eight vertical rows of rivets, while with bent corners there are only two rows, and if the stack was not very large, possibly only one. Remembering that the cost of a stack varies with the number of rivets, helps to make this clear.

Where possible, offsets or elbows in building stack should be omitted. These special parts are expensive, also they form places for collection of dust and soot, and usually rust out quicker than other parts. There are many other points which could be touched on, but I feel that to attempt to prolong this paper would defeat its purpose.

APPENDIX

TYPICAL SPECIFICATION FOR GUYED STACKS

The contractor for this material shall furnish and erect one steel stack, ... inches inside diameter by ...... feet high. The entire stack shall be made of mild steel plates, ... inch thick and of the quality known to the trade as tank steel.

Stack shall be built in courses not less than 60 in. high, which shall be so arranged that the bottom of each course telescopes into the top of the course below. The bottom shall be round with a ...... angle of the same thickness as the stack, or of special design as may be shown on plan.

If required there shall be provision for a smoke connection on the side, same to be of the same diameter as the stack, or if different shape to have approximately the same area. The top shall be reinforced with suitable angle ring. There shall be ...... 4-part guy bands made of ...... bar iron or steel, also ...... sets of ...... inch guy wires. Guys are to be properly set and fastened with suitable clips and turnbuckles. Anchorages for guys will be furnished by owner.

The stack shall be single riveted on all seams with ...... diameter rivets 3½ in. c/c., cone or steeple head rivets both in and out preferred. If field-riveted, rivets may be driven flat inside. There shall be a painters pulley and ...... feet of ¼ in. flexible wire furnished and set. Stack shall have one shop coat of silica graphite paint and also one field coat, after the erection is finally completed. Both coats outside only.

PRACTICAL NOTES ON GUYED STACKS

It has been the convention to make stacks heavier near the bottom than at the top; for instance, most specifications read, half ¼ in. and half $\frac{3}{16}$ in., or one-third each of ⅜ in., $\frac{5}{16}$ in. and ¼ in., etc. In guyed stacks this is unnecessary and poor engineering. Additional thickness at the bottom is not necessary from a viewpoint of strength, as there is no bending movement, there being merely the compression due to the weight of the stack which is negligible.

From a viewpoint of durability it is well known that stacks corrode near the top much faster than elsewhere, the point of condensation usually falling in the upper third. In other words it has been the practice to make the part most liable to corrosion the thinnest. If most guyed stacks were turned upside down, their life would be largely increased and the strength be unimpaired. For these reasons a moderate thickness the entire height is the most logical solution of the problem.

Guy bands are usually made of 2½x$\frac{5}{16}$ in. material for stacks 30 in. and smaller; 3x⅜ in. up to 42 in.; 4x4½ in. up to 5 ft. and heavier in proportion for the larger sizes.

Guy wire is usually ⅜ in. diameter up to 30 in. by 80 ft.; ½ in. diameter between 36 in. and 48 in. up to 80 ft. high; ⅝ in. diameter between 48 in. and 60 in. up to 100 ft. high, and heavier in proportion to diameter and height for the larger and higher sizes.

Stacks under 60 ft. usually have only one set of guys, stacks from 60 ft. to 80 ft. two sets, and stacks 80 ft. and up, three sets.

Rivets should be ⅜ in. for No. 10 gage steel, ½ in. for $\frac{3}{16}$ in. plate; ⅝ in. for ¼ in. and $\frac{5}{16}$ in. plate; ¾ in. for ⅜ in. plate, and up, spaced 3½ in. center to center for all sizes.

Hoods are undesirable and should be omitted unless the stack is to be out of commission for long periods. On short stacks they interfere with the draft.

Heavy cast-iron bases are unnecessary on guyed stacks as they serve no purpose which a concrete foundation does not serve. There is no load or strain on a base of this character other than the dead weight of the stack. If a cast-iron base is used the stack should not be rigidly attached or bolted to same, as a guyed stack must inevitably sway somewhat, which would only tend to break the base.

The arrangement of plates and rivet spacing given is that which is usual for ordinary stack work. Should it be necessary to insure that the stack will be streak proof, rivet spacing should be closer and all seams should be calked. Doing this increases the cost and is seldom worth the cost.

The durability of a steel stack depends largely on the care taken of it, and the use of proper paint. Ordinarily lead and oil paints do not stand the high temperatures to which stacks are subjected, invariably cracking and peeling. Graphite paint of a known quality has been found the best.

EXTERIOR OFFICE-BUILDING STACKS

The specifications for exterior office-building stacks are in general the same as for interior office-building stacks.

Stacks for use on the exterior of office buildings should preferably carry their weight to the ground, the sections being made with angle joints. Suitable bands should be placed frequently to maintain it in position. New York City building laws call for a band at every floor. Bands should permit free expansion and contraction of the stack. This class of stack should be made with the bottom of the upper courses telescoping into the top of the lower courses and further provision should be made to prevent leaks at the angle joints. The necessity for this is great as owing to the great height and the proportion to the comparatively small area of this type of stack, the condensation is excessive. If these precautions are omitted, the stack will be found to streak beautifully over practically its entire surface, thus becoming an eyesore and a source of much contention and trouble.

TYPICAL SPECIFICATIONS FOR INTERIOR OFFICE-BUILDING STACKS

The contractor for this material shall furnish and erect on steel stack, ... inches in diameter by ...... feet high. The entire stack shall be made of mild steel plates, ... inch thick and known to the trade as tank steel.

The stack shall be built in courses not less than 60 in. high, which shall be so arranged that the top of each course telescopes into the bottom of the course above. The stack shall be made in sections preferably 30 to 50 ft. long; this, however, is to be finally decided upon at the time the stack is built and must be arranged, so that the sections are not too long to handle into position at the site.

Each section must be supported from the steel work of the building (such supports being in the structural steel contract) by two suitable angle lugs, placed near the top of each section. Sections to be hung so that they lack one inch of touching with each other, the joints being covered by outside straps, 6 in. wide, attached to the lower course of each section so that it will slip 2½ in. over the top of the section underneath. The strap to be of the same thickness as the course to which it attaches.

The object of this arrangement is to provide for the expansion of the stack due to variations in temperature without strain to the stack, steel work or masonry.

The stack shall extend through the ceiling of the boiler room approximately one and one-half times the diameter and shall be arranged with a connection to receive the smoke breeching. The bottom shall be ...... in. thick with a clean-out door, and attached to the stack by a 3x3x...... outside angle ring.

Near the top, where the stack extends through the shaft, there shall be a hood provided, of the same thickness as the top section, which will extend 6 in. beyond the walls of the shaft and rest on same. The stack shall pass through this hood, but not be permanently attached to it. A loose joint packed with asbestos cement is preferred. The top of the stack shall be reinforced with a 2½x2½x$\frac{5}{16}$ in. angle ring placed on the outside.

There shall be one shop coat and one field coat of Dixon's silica graphite paint on the outside only.

PRACTICAL NOTES ON INTERIOR OFFICE-BUILDING STACKS

The remarks on thickness of stacks made under the head of guyed stacks apply also to office-building stacks. The most satisfactory thicknesses are ¼ in. or $\frac{5}{16}$ in. The New York Building Department will not permit stacks to be built less than ¼ in. thick unless they are galvanized sheets.

The size of rivets and spacing should be the same as mentioned previously. Sometimes it is preferred to use oval or rectangular stacks. The same general specifications would still apply.

Avoid using angle corners in rectangular stacks; the construction is much more expensive, without having any advantages. Rectangular stacks should preferably be built in courses about the same height as for round stacks, corners to be bent to 3 in. radius. Each course should be in two parts (i. e., two vertical seams), unless the stack is very large when four parts may be necessary.

Sometimes, it is not feasible to carry the stack on the steel work of the building; in such a case the stack should extend to the boiler room floor, where suitable foundations should be prepared. Sections should be made with 3x3 in. angle rings of the same thickness as the stack, so that all the weight is transmitted to the base. The angle ring at the bottom should be ⅛ in. thicker than the stack, but need not be larger than 4x4 in. This type of stack needs to have braces to insure it staying in position and the braces must permit free expansion. Braces every other floor are close enough.

PRACTICAL NOTES ON SELF-SUPPORTING STACKS

The specifications and remarks on the other types of stacks apply very generally to self-supporting stacks. To write a satisfactory specification, which would give all necessary detail information, would require a treatise on the subject which is beyond the intention of this appendix.

To properly design a stack of this order is a problem for a mechanical or structural engineer to work out at length and should not be attempted by off-hand specifying. For intelligent bidding, a properly designed drawing of the stack required should be prepared.

For those interested, the following practical notes are given as guides, but for the theory of stresses in structures of this and similar types, we refer to the technical literature on the subject.

Self-supporting stacks should preferably be not over 25 times their diameter in height. Heights greater than this cause excessive stresses. Bases should flare to at least one and one-half times the diameter, tapering to meet the straight part of the stack at approximately one-sixth of the height from the bottom. Thicknesses at the bottom hardly ever need to exceed ½ in. as far as strength is concerned.

Generally speaking, for stacks up to 8 ft. in diameter and say 150 ft. high, with a normally flared base, ⅜ in. is thick enough for the lower part. No part should be less than ¼ in. thick, preferably $\frac{5}{16}$ in. thick if durability and final cost is to be considered.

As self-supporting stacks are usually intended for permanent use and also are usually quite large, a ladder, extending full height, also a painter's track and trolley, should be specified. If the stack is to be lined, the lining should be divided into sections say 20 ft. in height, and each section carried by a suitable interior angle ring. This arrangement provides for the expansion of the lining. Further, the lining should not touch the shell of the stack. It is the custom to leave a space of 1 in. all the way around, filling the same with loam.

If openings are cut into the side for smoke connection, they must be properly reinforced. In self-supporting stacks this is most important, as the lower part is under severe strain. In guyed stacks this condition does not exist. A clean-cut door should be provided.

See final paragraph of notes on guyed stacks for paint.

No. 586

SOME COMPARATIVE TESTS OF SIXTEEN-INCH ROOF VENTILATORS

By H. L. Dryden, W. F. Stutz, and R. H. Heald,[1] Washington, D. C.
Non-Members

DURING the summer of 1920 a study of some fifty 16-in. ventilators, was made at the Bureau of Standards, Washington, D. C., the ventilators having been submitted by the Construction Division, Q. M. Corps, of the Army. The following paper gives a brief summary of some of the results of the study.

The factors affecting the performance of a ventilator and the things which must be taken into consideration in the choice of a ventilator are so numerous that it was impossible to attempt a complete study. We limited ourselves definitely to certain specific phases of the problem and it must be kept in mind that the tests about to be described are particular tests with a certain experimental arrangement. The question as to how far the results of these tests apply to any other arrangement is left open.

Our experiments were confined to questions concerning the volume of air exhausted per minute by the ventilators, which is dependent upon many factors. For example, if the air does not have free access to the room, little air will be exhausted. If there are obstructions near the ventilator, the performance will be affected. The most important factors, however, affecting the performance of a ventilator are (1) the difference in temperature between the air in the room and the air outside and (2) the speed of the wind blowing across the top of the ventilator. Our experiments were restricted to these two factors.

The effect of a temperature difference is to produce the familiar chimney action, an action common to all ventilators, including an open pipe. The design of the ventilator affects the amount of air exhausted under a given temperature difference only in-so-far as more or less resistance is offered to the flow of air. If the ventilator passage is obstructed, less air will be exhausted. From this standpoint, a straight open vertical pipe is the ideal ventilator, but considerations of weatherproofness prohibit its use.

The exhaust due to the wind depends primarily upon the design of the ventilator. Our first experiments were arranged so that the tempera-

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ture at the entrance and exhaust were the same, so that there was no chimney action. The exhaust for various wind velocities was measured with the set-up shown in Fig. 1. The wind was produced by one of the wind tunnels of the Bureau of Standards. For the purpose of these tests the exhaust fan of the tunnel was removed and a blower fan substituted. By means of suitable honey-combs the velocity was made as nearly uniform as possible across the stream. The ventilators were placed in front of the mouth of the tunnel on the end of a vertical pipe, the wind stream being horizontal. A horizontal pipe containing the measuring apparatus was joined to this vertical pipe by means of an elbow.

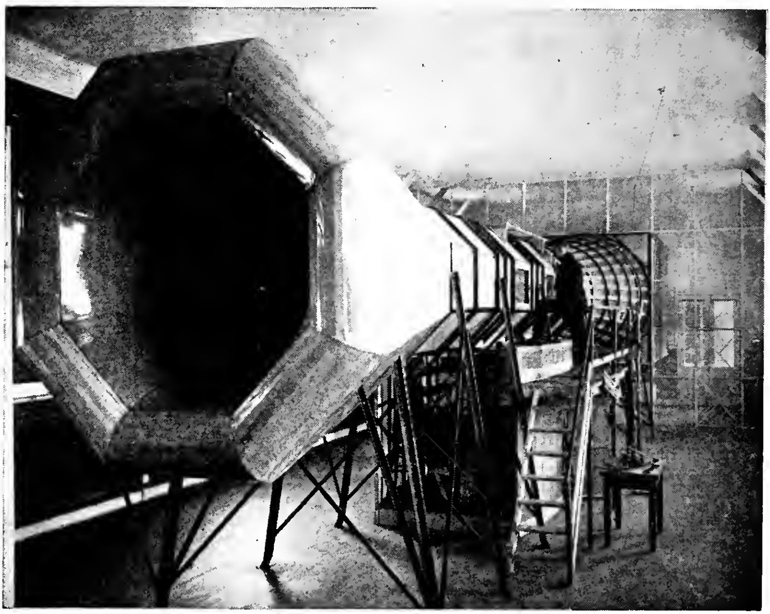

FIG. 1. WIND TUNNEL OF THE BUREAU OF STANDARDS AS ARRANGED FOR VENTILATOR TESTS

DETERMINING WIND SPEED

The speed of the wind was obtained from the readings of a tachometer connected to the shaft of the wind tunnel motor, the readings of the tachometer having been previously standardized in terms of an anemometer placed in the position later occupied by the ventilators.

The volume of air exhausted was obtained from the deflections of a small wire suspended freely in the horizontal pipe from a watch-bearing mounting. The wire anemometer was calibrated by comparison with an orifice meter.

Measurements were made of the volume of air exhausted per minute by the ventilators at wind speeds of 4, 8 and 12 miles per hour. For comparison, index numbers or wind ratings were obtained by expressing the volume exhausted by a ventilator as a percentage of the volume exhausted through the open pipe in the same time at the same wind speed. The wind rating of the open pipe is therefore 100 at all wind speeds. The accuracy of the measurements is about 5 per cent, and smaller differences

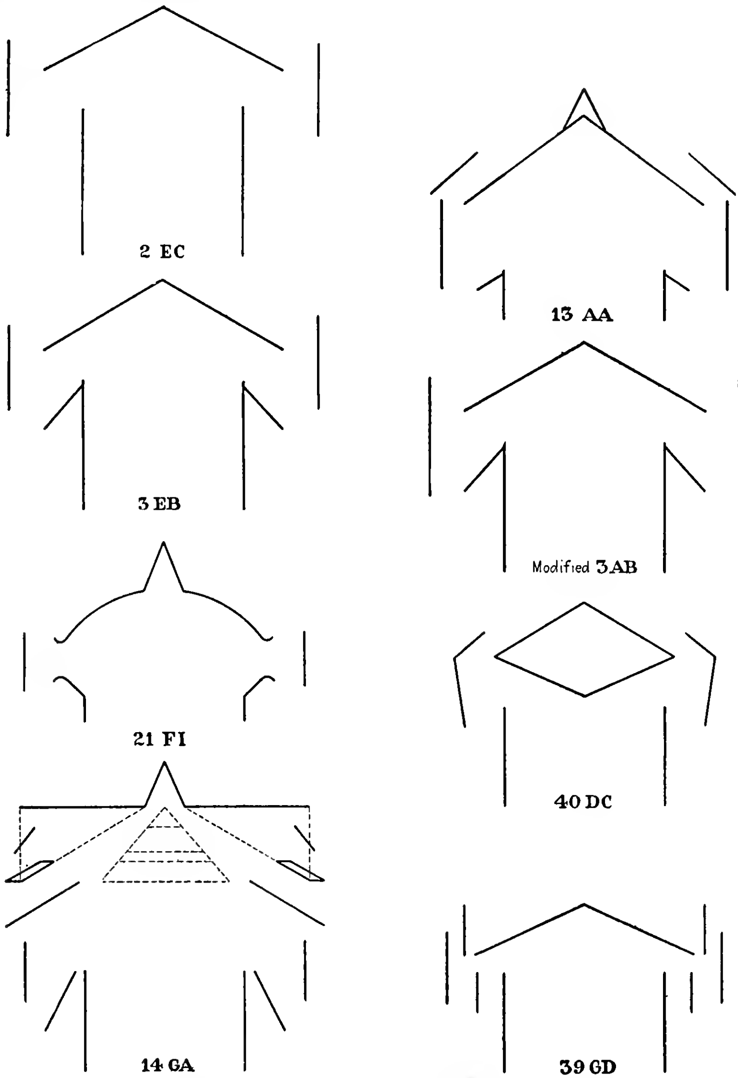

FIG. 2. SPECIMEN TYPES OF VENTILATORS TESTED.

in ratings are of no significance. Some ventilators exhaust less air than an open pipe, some more; the best ventilators have a rating of 150, the exhaust being one and one-half times as much as from an open pipe. The exhaust of an open pipe with the set-up used was about 230 cu. ft. per min. for a wind speed of 10 miles per hr.

In addition to the measurements of the volume of air exhausted at varying wind speeds with no temperature difference, measurements were made of the flow of air through the ventilators with a given difference in pressure in order to simulate the effect of a difference in temperature

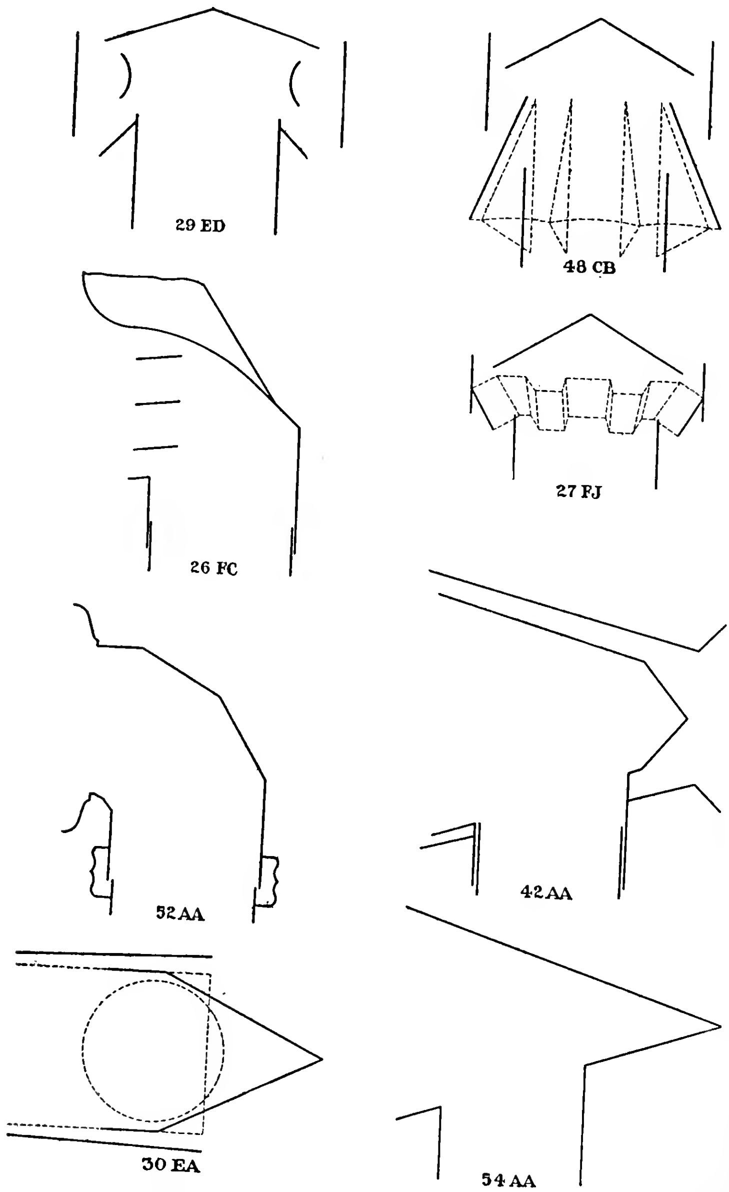

FIG. 3. SPECIMEN TYPES OF VENTILATORS TESTED.

between the air of a room and the air outside. An electric fan was placed at the entrance of the ventilator pipe line and the volume of air per minute passing through the pipe was measured at various speeds of the fan. It is evident that if the fan is running at a uniform speed and the same flow is obtained with two different ventilators, the resistances of the two ventilators are equal and consequently under a given temperature difference the same volume of air per minute would be exhausted by the two ventilators. If the flow through one ventilator is less, its resistance is greater and it would exhaust less air under the same temperature difference. Ratings were again made on the basis of the open pipe as 100. In these resistance ratings, 100 is the maximum attainable, since the resistance of the ventilator is added to that of the pipe.

VENTILATORS DESCRIBED

Ventilators may readily be divided into those of stationary and rotary forms, but any further sub-division for purposes of discussion is difficult. The simplest type of ventilator consists of a cap over the top of an open pipe with a band around pipe and cap to keep the rain from beating in. Such a ventilator exhausted 94 per cent as much air as an open pipe at the same wind velocity. In other words, it has a wind rating of 94. It permitted 86 per cent as much air to pass as an open pipe in the second experiment. This will be expressed by saying that the *resistance* rating is 86.

A simple modification of No. 2 is shown in ventilator No. 3 (Fig. 2) where a lip is placed on the pipe. This ventilator is very sensitive as regards its orientation relative to the wind. On introducing a smoke stream it was found that the air passing under the band separates into two parts, one part passing under the cap and diagonally out at the sides of the ventilator, the other passing underneath the lip and around the pipe. The quantity of air going by the two routes depends on the orientation of the ventilator relative to the air stream. The results varied by about 10 per cent in different experiments, according to the orientation. This ventilator had a wind rating of 96 and a resistance rating of 88.

EFFECT OF BAND

To investigate more fully the effect of the band, the band was lengthened as shown in modified No. 3 (Fig. 2) so as to extend a little below the lip. The wind rating increased from 96 to 130 and on an examination with a smoke stream it was found that no air entered the ventilator at all. The air blowing on the ventilator passed underneath the lip, the ventilator exhausting all the way around. Ventilator No. 13 (Fig. 2), of somewhat similar construction, but of different proportions, had a wind rating 138 and a resistance rating 95.

Fig. 3 shows other modifications, some complicated and some simple. No. 21 illustrates the effect of making the openings in the ventilator too small for the air to pass out freely. Its resistance rating was 56, its wind rating 85. No. 14 illustrates a very complicated construction of low resistance, resistance rating 100, wind rating 73. No. 39 is another complicated one, resistance rating 75, wind rating 77. In the case of No. 29, extending the band below the lip increased the wind rating from 91 to 113. Its resistance rating was 78. No. 48 is better than the simple types No. 2 and No. 3, but not as good as modified No. 3, its wind rating being 109 and its resistance rating 93. No. 27 is another ventilator with small exit passage for the air, its resistance rating being 50 and its wind rating 80.

These ventilators are typical of the stationary ones and the surprising fact is that the best exhaust is obtained with a very simple construction.

No. 26 (Fig. 3) is a simple type of rotary ventilator consisting of an elbow with a wind vane to hold the opening away from the wind. Its wind rating was 87, resistance rating 83. Its resistance rating is low because of some large damper supports in the pipe which are not shown in the sketch.

Fig. 3 illustrates other rotary ventilators. No. 52 differs from No. 26 in that the pipe is free from obstruction and the air is deflected outward by means of a lip on the elbow. Its wind rating was 150, its resistance rating 95. No. 30 is another type in which air is permitted to pass through a passage in the ventilator. Its wind rating was 91. On stopping up the passage, the wind rating was increased to 135, so that the passage way is detrimental to the performance of this particular ventilator. No. 42 also has a passageway for the air. Stopping up the passage had no effect on the wind rating, the rating remaining 149. As a result of this observation we designed the simple form No. 54. This cone type had a wind rating 149, the same as No. 42.

CONCLUSIONS

In conclusion, two points should be emphasized. The first is that no general statement can be made as to the relative merits of rotary and stationary, or mushroom and siphon, ventilators. The performance depends on the particular models. It is possible to build a good stationary ventilator as well as a good rotary ventilator, and there are poor ventilators of each type. The second point is that the most effective way of obtaining a large volume of air exhaust is by making use of the region of low pressure produced at the back of a properly designed obstacle. It is best not to allow the air to enter the ventilator for it must then be exhausted and will be exhausted at the expense of the air in the ventilator pipe.

The detailed results of the investigation are being prepared for publication by the Bureau of Standards as a technologic paper. We desire to acknowledge the efficient assistance of Messrs. R. D. Campbell and M. Temin in connection with the tests.

DISCUSSION

S. H. TAYLOR (written): Although I have been manufacturing roof ventilators in a large way for very many years, I confess that I am unable to thoroughly understand some of the statements in this paper and some of the technical terms used. I would, therefore, greatly appreciate some clear explanation of the terms used and performances of some of these ventilators under these tests.

I understand that in July, 1918, a test of 33 16-in. ventilators was made at the U. S. Bureau of Standards and the results obtained showed the air displacement in cubic feet per minute, with a wind velocity of varying speeds per hour.

I would like to ascertain whether the expression, "Wind Rating," frequently used in your paper means and is, air displacement or air exhaust, and would one ventilator with a wind rating of say 149 or 150 have greater air exhaust or air displacement than another ventilator with a wind rating of 100? Also what relation has wind rating and resistance rating to the actual efficiency of the ventilator?

I observe that Fig. 13AA has a wind rating of 138 and a resistance rating of 95; Fig. modified—3AB, has a wind rating of 130 and no resistance rating specified; Fig. 52AA, has a wind rating of 150 and resistance rating of 95; Fig. 42AA, has a wind rating of 149 and no resistance rating specified; Fig. 48CB, has a wind rating of 109 and resistance rating of 93. Can you tell me which of these ventilators is the best? In other words, which will displace the most cu. ft. of air per min. at some uniform speed?

I do not understand the resistance data. For example, it is stated "if ventilator passage is obstructed, less air will be exhausted." Further on, it is stated that Fig. 52, *with the pipe free from obstruction,* had a resistance rating of 95 and wind rating of 150; whereas Fig. 30 with an unobstructed passage had a wind rating of 91, but when the passage was stopped up, the wind rating was increased 135. The paper states that the passage way is detrimental to the performance of this ventilator, although it distinctly states that when the ventilator passage is obstructed less air will be exhausted. Again it states that stopping up the passage of ventilator, Fig. 42, had no effect on the wind rating.

I assume that measurements of volume of air exhausted at varying wind speeds with *no* temperature difference was, or could be, called test No. 1 and that when the measurements were made of the flow of air through the ventilators with a given difference in pressure, were known, or could be, called test No. 2 and indicated resistance ratings. In these resistance ratings it is stated that 100 is the maximum.

It further states that if the flow of air through the ventilator is less its resistance is greater and it would exhaust less air under the same temperature difference.

I would, therefore, understand, for instance, that ventilator Figs. 13AA and 52AA with a resistance rating of 95, would keep the air from being displaced or exhausted to a greater extent than say ventilator Fig. 48CB.

I also wish to ask why, in reference to Fig. 14, showing a ventilator of a *very complicated construction,* it states that said ventilator has a low resistance rating of 100 and in other parts of the article reference is made to *ventilators of the most simple construction being the best.*

Again in commenting on Fig. 26, it refers to resistance rating of 83 as low. While in referring to the simplest type of ventilator, consisting of a cap over the top of an open pipe with a band around the pipe and cap to keep the rain from beating in, it is stated that said ventilator exhausted 91 per cent as much air as an open pipe at the same wind velocity. Whereas, later it is stated that said ventilator permitted 86 per cent as much air to pass as an open pipe. The wind rating was 94 and resistance rating 86.

H. L. DRYDEN (written): The quantity of air exhausted by a ventilator depends essentially upon the installation conditions, i.e., the air-tightness of the room, the position of the ventilator in the room, the length of pipe attached, etc. For any particular installation the volume exhausted depends on the difference in temperature between the air inside the room and the air outside and on the velocity of the wind. For this reason figures for the capacity of a ventilator are meaningless unless the particular set-up, the temperature difference and the wind speed, *all three,* are given. Two distinct tests were made with our installation. The first test was of the volume of air exhausted at varying wind speeds when there was no temperature difference and shows the value of the ventilator as an ejector. The second test was of the volume of air exhausted when there was no wind and a given pressure difference was applied so as to simulate the effect of a temperature difference. This test shows the value of the ventilator as a chimney. An attempt was made to reduce the three variables to two by choosing as a standard of reference a straight, open, vertical pipe and by expressing for this particular set-up the results in the form of ratios of the observed exhaust to that of the open pipe under the same conditions. It seems reasonable to believe that these ratios will not vary widely in installations which do not differ markedly from this installation.

Thus the "wind rating" is the ratio of the air exhausted by the ventilator to that exhausted by the standard straight, open, vertical pipe when there is no temperature difference and there is a wind, and the "resistance rating" is the ratio of the air exhausted by the ventilator to that exhausted by the open pipe when there is no wind and there is a constant pressure difference applied. It was found that the "wind rating" was very nearly independent of wind speed and the "resistance rating" nearly independent of pressure difference.

It follows that the ventilator with the highest wind rating is the best for an installation where there is no temperature difference, and the one with the highest resistance rating is the best where there is no wind. For all-around use, where there is wind and also a temperature difference, the ventilator with both the ratings high is to be preferred.

No. 587

REPORT OF COMMITTEE ON RESEARCH

To the Members of the Society:

This Committee has held seven meetings during the calendar year. Two of them were Executive Committee meetings; the others were either meetings of the whole Committee or joint meetings with the Council of the Society. The attendance has been unusually good, considering it is composed of busy men, scattered about the country, some of whom had to travel long distances to attend the meetings.

Every meeting has brought forth enthusiastic interest in the present and future progress of the Laboratory. Most careful attention has been given to every detail by those present. An earnest effort has been made to further the best interests of the Society, the contributors to the Research funds and the public generally keeping in mind that no special or selfish interests are to be served, either inside or outside of the Society.

It is sometimes difficult to arrive at a definite conclusion as to what should or should not be undertaken or reviewed by the Laboratory. Almost anything that has to do with the physical well being of humanity or the economics of the healthful comfort and convenience of mankind are prime objectives of the Laboratory.

As this becomes more generally understood and our facilities become more generally known, we will most assuredly be appealed to with greater frequency and insistence to undertake investigations which may not at that time be of the widest general interest nor entirely divorced from individual gain, but which will form a link in the chain of reliable information that must ultimately be available in order that we may have such a comprehensive knowledge of the art and science of heating and ventilation as will make our findings and recommendations authoritative and incontrovertible.

As an example, an appeal has been made by one of the largest corporations in this country, to outline and supervise some investigations it proposes making to determine the heat losses from differing types of mechanics homes and constructed of various materials, in an endeavor to learn which of the methods now in use for heating residences are the most economical to operate. The end in view is to adopt a standard plan, a standard form of construction and a standard method of heating such places; then to proceed with the building of thousands of such houses for miners and mill employes.

Still another project has been undertaken by the Board of Education of the City of Minneapolis. Rooms have been equipped with several different types of standard heating and ventilating apparatus and the Research Laboratory has been requested to outline the method of procedure and to observe and direct the experiments.

In deciding to undertake either the actual work or the supervision of these projects, the Committee has weighed carefully what can be gained for the Society and for the public. Then it has considered what value will the data be when a future project arises, or what value will it have as forming the ground work for some other, and perhaps a more exhaustive investigation.

No effort has been spared to act with strictest impartiality where there has been a likelihood of a collision of opinions. Injury to the reputation of an individual, to a corporation or to the products of manufacturer is a thing to be avoided as much as possible; but when the greatest good to the greatest number requires a report from the Laboratory, the findings of which on any particular subject make comparisons which are not altogether to the liking of some particular group of persons or corporations, the Laboratory or the Committee is likely to be criticized.

To avoid this kind of criticism as much as possible, we would recommend that published reports on incomplete work be in the form of reports of progress, eliminating harmful details. When the work is finally completed, then all of us should stand manfully together and insist on publishing all details, regardless of the consequences. In no other way will the universal confidence of the public be attained.

Such a situation has not yet arrived, but it is plainly on its way; hence it seems as though now is a good time to outline the future policy of the Committee so there can be no misunderstanding when it does arrive.

The Directors' report covers all the details regarding the laboratory, equipment, personnel, the work under way and in prospect, inside and outside the laboratory; articles already published and in preparation; visits to Chapters and to colleges and work now under way and to be started, facilities available and equipment needed by the universities willing to work with the Laboratory.

The Treasurer's report will cover the funds subscribed, the expenses and the amount available. The Committee thanks the members for their efforts to raise funds for the Research Laboratory and hopes for a continuance of these same splendid efforts. It will require vigorous and organized effort by the entire membership to get subscriptions in amounts large enough to properly sustain the Laboratory.

The work has now arrived at the point where the preliminary preparation will begin to show in definite results. With work being conducted by a number of universities, great progress should be made in about another year. Some of the work is naturally slow and tedious; then again some of it will take years to accomplish. When we stop to consider the fact that we are dealing with some problems which have been investigated without solution for centuries, we must not expect them to be solved by our Laboratory in a few months.

A most unfortunate and appalling blow was inflicted upon us by the sudden death of Dr. John R. Allen, whom we all respected and admired so much, not only for his high professional standing but for his many charming personal qualities. His whole heart was wrapped up in the research work being conducted by the Laboratory and beyond question of a doubt, his sudden death was largely a result of a run down physical condition due to almost constant travelling about, with the consequent exposure and inability to properly take care of himself while away from home.

We have not yet selected his successor. He had engaged Dean L. A. Scipio of Robert College, Constantinople, to join the staff at the Laboratory as his assistant. He arrived the day Director Allen died. We engaged him later to take up the work as Acting Director for the remainder of the fiscal year, pending his determination to remain in this country or to go back to Constantinople; this arrangement also affords us an opportunity to better measure his ability to carry on the work.

He was highly esteemed by Director Allen; has had just the training needed for a director; is a Christian gentleman of quiet refinement; has an investigative turn of mind and has had that broad experience in practical business affairs which is so necessary to give a sane and safe direction to the affairs of such an institution. He has been spending most of his time the past two months visiting the universities to canvass their facilities, equipment and desires to cooperate with us.

Words fail to express the appreciation of the Committee for the generous and whole-hearted support given by Professor Hood and his excellent staff of assistants at the Bureau of Mines. Nothing is too much trouble for them and any work undertaken by our staff always brings to hand proffers of all the space and assistance we can make use of.

A large measure of the responsibility for the management of the business of the Laboratory has been left entirely with the Chairman of the Committee. If anything has been done to which exception can be taken it has been done from lack of understanding of what represented the wishes of the majority, also a desire to keep things progressing, rather than a desire to dominate. The confidence which has been reposed in the Chairman by the entire Society is considered by him as most remarkable and has been a constant stimulus to the exercising of greater care in the handling of all matters on which his opinion was sought or directions were asked for, than such matters might otherwise have received, to avoid doing anything which might embarrass those who were responsible for his selection.

It has been a pleasure to serve on the Committee, and it is hoped the next Chairman may be favored with the same confidence and cooperation as it was the good fortune of the last one to receive. The Laboratory now seems to be quite firmly established and its importance will grow with its accomplishments. It has wonderful opportunities ahead. That it may fulfill every one of the objects for which it was founded and to the everlasting credit of all those who worked so hard and so long to establish it, is the invocation of this Committee.

F. R. STILL, *Chairman.*

APPENDIX I

TO THE COMMITTEE ON RESEARCH:

Offices and Laboratories. The offices and laboratories are located at the Bureau of Mines Experiment Station, Pittsburgh, Pa. In addition to the three rooms occupied last year, another has been added so that we now have numbers 282, 281, 279, 277. The first room is used as an instrument and work room and the remaining three are used as the business offices. The constant temperature room has been completed in the basement of the building and we expect to have the refrigerating machine installed within a short time.

Personnel. The staff now consists of Director L. A. Scipio, F. C. Houghten, physicist and specialist in heat transmission, O. W. Armspach, in charge of ventilation work, Louis Ebin, engineering assistant, and Elinor E. Mellon, secretary.

Equipment. The inventory of the Laboratory is appended and consists of Bayonet immersion heaters, potentiometer indicators, resistance bulbs, resistance thermometers, two Peterson Palmquist Gas Analysis machines, an electric boiler, cold room, etc. Numerous pieces of apparatus have been donated by various manufacturers, the most prominent of which are the Taylor Instrument Co., American Blower Co., Atmospheric Conditioning Corp., and the Armstrong Cork Co. The former company has generously given high grade instruments aggregating a value of about $600 and the last mentioned company lined our constant temperature room with cork and installed the refrigerator door. The total inventory aggregates to the sum of $4,212.56.

Bibliographical Work. The card index which had already been built up has been considerably augmented, principally through the courtesy of the Journal of Industrial Hygiene which very kindly allowed us to take copies of their bibliographies on such subjects as dust, temperature effects on the body, ventilation, and amount of moisture excreted by the body. We are continually adding to our bibliography as the current magazines come in, and it is our idea to make it as extensive as possible.

Standards Already Adopted. At the time of the last report, the Laboratory had endorsed the following standards which had already been adopted by the Society:

Report of Special Committee on Standards for Flanged Fittings and Flanges, printed in the TRANSACTIONS, Vol. XVIII, 1912, p. 44. This report adopts the standards of the *A. S. M. E.*

Report of the Committee on the Use of the Pitot Tube, printed in the TRANSACTIONS, Vol. XX, 1914, p. 210.

Report of the Committee on Code for Testing Heating Boilers as revised in 1919. This report may be improved by a statement in the table, of results which shows after each result, the method of computation, similar to the statement made in the *A. S. M. E.* standard method of testing boilers.

The only additional standard endorsed by the Laboratory is the Synthetic Air Chart by Dr. E. V. Hill as a standard for ventilation, adopted by the Society at the Semi-Annual Meeting held in St. Louis, June, 1920.

RESEARCH WORK OUTSIDE THE BUREAU

The late Director Allen attached considerable importance to the work being carried on by the various universities in cooperation with the Research Laboratory. The following work may be reported as in progress:

The University of Michigan is making an examination of the manner in which heat is lost from steam pipes buried in the ground. It is just possible that this work will be transferred from the open to the laboratory in order to better control the various elements concerned. Professor Allen arranged to contribute $300 per year for this work but to date none of this money has been used.

The University of Minnesota in cooperation with this Laboratory is investigating the radiant heat lost from the various types of steam radiators, and the work is now in progress.

Pennsylvania State College is determining the heat losses from building materials by the electrical plate method. The apparatus was constructed under the direction of the Bureau and the tests are being conducted by Mr. Houghten of the Laboratory. The project will require a maximum of $500 of the Society's funds, and about $100 has been spent on this work at the present time.

As is generally well known, the Engineering Experiment Station at the University of Illinois has for three years been conducting extensive experiments on warm-air furnaces under the direction of Professor Willard. Most excellent results have been obtained and the Laboratory is keeping in touch with the work. Owing to the nature of their arrangement with the Warm-Air Furnace Association, this work cannot well be carried on cooperatively in the fullest sense, but the most cordial relations exist between the Station and the Research Laboratory, and we will no doubt profit extensively thereby.

At the University of Ohio, Professor Magruder is in the midst of preparation for carrying on research in connection with gas-burning grates and ranges, and this is to be done in cooperation with the Research Laboratory.

At Armour Institute, Professor Peebles has been conducting experiments on heat losses through building materials for several years, and arrangement has been made to cooperate with them, and we may expect definite results in the way of a paper within the next few months. This institution bids fair to be one of the most active of any of our list.

At Ohio Northern University, arrangement has been made for carrying on a comparative study between the direct radiation system of heating and the warm air plenum system. This is to be done in connection with two buildings of similar construction and heated in the different ways. It may also be possible to make some discoveries as to the heat losses through windows.

Professor Dibble of Carnegie Institute of Technology is constructing chimneys of different kinds, such as are found in residences, and is carrying on tests with a view to ascertaining their efficiencies.

The manager of the Morgan Park Development Co., Duluth, Minn., has expressed his desire to cooperate with us in the study of heat losses through building materials by the construction of six small houses of different materials, the idea being to maintain the same temperature in each, electrically, and thus determine the amount of energy required in each case. He is submitting plans and asking for suggestions from the Laboratory. We also hope to be able to do some work on infiltration in connection with this experiment.

Negotiations are under way for a cooperative arrangement with McGill University. This institution has already been conducting experiments on heat losses through windows, and in addition to continuing this work, we expect that they will assist us in our work on infiltration.

Arrangements have also been made with the American District Steam Co. to conduct some tests on air space coverings used on underground pipes, but no work has been started.

Rensselaer Polytechnic Institute has undertaken to determine the temperature gradient through glass. The apparatus for making these tests was designed by the Laboratory and constructed at the institute. These tests will have a fundamental bearing upon our ideas of heat transmission. This institution has also agreed to make a study of the air conditions and other ventilation problems in some of their class rooms.

The Laboratory has arranged to cooperate with the Minneapolis Board of Education in conducting ventilation tests in their Whitney School. This school is a six-room building with a gymnasium and office and each class room is to be equipped with a different ventilating outfit with a view of determining the qualifications of the different systems. The following outfits are to be used:

1. One room equipped with the Beery System;
2. One room equipped with the Wheeler Open Window System;
3. One room equipped with the Moline Heat "Uni-Vent" System;
4. One room equipped with the St. Louis "Ozonator" System.

Professor Allen made arrangements to have Mr. Armspach go to Minneapolis when the installation was ready and assist in conducting the tests. These tests will probably have considerable bearing on the future ventilation of school buildings and it seems desirable that Professor Allen's plans in this matter be carried out.

RESEARCH WORK OUTSIDE THE LABORATORY (CONTEMPLATED)

In addition to the work already in progress as indicated above, the following were contemplated by Dean Allen:

UNIVERSITY OF MINNESOTA: Prof. F. B. Rowley.
  Heat losses from building materials.
  Effect of humidity and temperature on the human system.

UNIVERSITY OF ILLINOIS: Prof. A. C. Willard.
  Determination of critical velocities in steam risers and lateral vacuum traps.

UNIVERSITY OF MICHIGAN: Prof. J. E. Emsweiler.
  Determination of flow in natural hot-water systems.
  Building material.

NEW YORK UNIVERSITY: Prof. Snow and Prof. Bliss.
  Pipe Sizes.
  Friction of flow in vacuum return lines.
  Vacuum pump capacities.

YALE UNIVERSITY:
  Test of direct radiation (electrical testing).
  Test of vacuum valves.

KENTUCKY STATE UNIVERSITY:
  Fractional valve capacities.
  Size of orifice at standard pressure.

JOHNS HOPKINS UNIVERSITY:
  Tests of special types of radiators:
    Hot-water steam heated.
    Gas radiator.
    Electrically heated.

UNIVERSITY OF WISCONSIN:
  Heat losses through pipe coverings.

RENSSELAER POLYTECHNIC INSTITUTE:
  Electrical method of testing radiators.
  Standard of ventilation.

The Sub-Committee on Subjects from the Executive Committee is planning to make new assignments, and as soon as this is done, other problems will be distributed among the schools with which the cooperative arrangement is being established. The idea is to assign the same problem to from three to five different institutions, having them work on it simultaneously in as far as possible under a plan which has been previously agreed upon between them and the Laboratory. In this way, the Laboratory will have a check on all of the work that is going on, and the final results ought to be more reliable than if they came from an individual institution.

Research Work at the Laboratory. A constant temperature room has been constructed for the purpose of carrying on a wide range of experiments both in the testing of radiators and the losses of heat through building materials, glass, etc. As soon as a refrigerating machine can be installed the work will begin on this actively.

The Bureau of Mines, in cooperation with the Research Laboratory is carrying on a series of experiments on air dust measurements and the calibration of the various types of dust counters. A determination of the efficiency of the water spray type of counters and the sugar tube has already been completed. Sugar tubes of high efficiency have been evolved. At present a comparison of the Hill Dust Counter, the Konimeter, and the Sugar Tube and the subsequent microscopic examinations is being made. This work will finally result in the adoption of certain calibration factors so that all dust determinations will be on the same basis regardless of the instrument used.

It is also proposed to investigate the dustiness of the air at the stone works in the Barre, Vermont, vicinity using the Sugar Tube, the Hill Dust Counter, and the Konimeter. The samples have been taken and the results will be available within a few weeks.

A theory of falling dust has been formulated and published in the December issue of the JOURNAL.

The tests to determine the quantity of air infiltrated through various types of constructions have been started and several tests have been made. Mr. Armspach of the Laboratory, has spent several months in perfecting a method of determining the amount of air entering a room by infiltration. This is a very delicate problem and has required a great deal of painstaking care and experimentation. Up to the present time, a number of tests have been made and the results have been quite satisfactory. A progress report showing the method of development is appearing in the current number of the JOURNAL, and which should be of great interest to all the members. These experiments when completed will establish limits to the quantity of air supplied in naturally ventilated buildings and together with the heat transmission tests being conducted at Penn State College, will form the basis of revised formulæ for figuring direct radiation. They are therefore of the utmost importance from a standpoint of both heating and ventilation.

An electrical boiler has been constructed with which to test radiators and the motor switch uesd in connection with this apparatus has been designed and is now being constructed in the shops of the Bureau of Mines.

John Blizard of the Bureau of Mines, in conducting boiler tests, is using the *A. S. H. & V. E. Code.* He has already written his statement of constructive criticisms on this Code.

Publications. Up to the present time, seven articles have been published in the JOURNAL originating in the Research Laboratory. They are as follows:

Investigations of Direct Radiation, by John R. Allen, January 1920.
Determinations of Radiant Heat given off by a Direct Radiator, by F. B. Rowley and John R. Allen, March 1920.
Heat Losses from Pipes Buried in the Ground, by John R. Allen, May 1920.
Relation of the Wet Bulb Temperature to Health, by O. W. Armspach, May 1920.
Continuation of Article on Heat Losses from Pipes Buried in the Ground, by John R. Allen, September 1920.
Modus Operandi of the Synthetic Air Chart, by John R. Allen, September 1920.
Theory of Dust Action, by O. W. Armspach, December 1920.

The following articles are in preparation and will appear later:

A Study of the Infiltration of Air in Buildings, by O. W. Armspach.
A Progress Report, January 1921.

Visits to Chapters. It is the intention of the Director of the Laboratory to visit the various Chapters of the Society, and in this way become acquainted with its members and arouse as much interest and enthusiasm as possible.

Visits to Colleges. During the last two months, some 30 colleges and universities have been visited with a view to ascertaining what their facilities are for conducting experiments, also the disposition of those in charge. The results have been most gratifying. In 75 per cent of the cases a willingness to cooperate has been expressed and frequently requests have been made to assign the work as soon as possible in order that it can be gotten under way. It is the intention to continue these visits in as far as time will permit, and thus increase our range of activity just as fast as it can be done.

Finances. Attached to this report is a list of the assets of the Laboratory and a budget report of the Laboratory from August 1, 1920 to January 1, 1921.

One of the purposes of the Laboratory is to be as useful to the members of the Society as is possible, and it is hoped that they will not hesitate to write in for such information as they need. In its early stages it may not be possible for us to answer every inquiry that comes to us but these inquiries will show us the lines along which we should prepare ourselves, and every courtesy will be shown to inquirers. One thing that should be mentioned, however, is that it is not the policy of the Laboratory to conduct tests on various pieces of apparatus and equipment which would in any way be of a competitive nature. If there is a new principle or something of especial interest to be gotten from such tests, then no doubt the Committee would see fit to assign such work for investigation.

At a recent meeting of the Executive Committee at the Bureau of Mines, it was decided to recast such results as have been obtained at the Laboratory in experiments already made, putting them in as simple and usable form as possible in order that they will serve the greatest needs of the practical man. This work has already been begun.

In his visits to the various colleges, the Director finds that the greatest need is for men who are competent to carry on the investigations. He is frequently told by men in charge of the laboratories that there is plenty of equipment and sources of advice, but that men for actually carrying on the observations are scarce. The greatest need at the present moment is to have more men on the staff. If it were possible to have a fund of from $30,000 to $40,000 to expend, the usefulness of the Bureau would be many times greater than what it can be at the present time with the present amount of money available.

Respectfully submitted,

L. A. SCIPIO, *Acting Director,*

A. S. H. & V. E. RESEARCH LABORATORY.

APPENDIX II

BUDGET REPORT OF A. S. H. & V. E. RESEARCH LABORATORY

AUGUST 1, 1920—JANUARY 1, 1921

	Allowance	Spent	Remaining
Salary of Director and assistants........	$17,120.00	$5,889.00	$11,231.00
Office supplies and expense..............	600.00	435.04	164.96
Traveling expense	1,500.00	887.99	612.01
Expense for Heat Investigation	1,000.00	15.32	984.68
Expense for Ventilation Investigation ..	1,000.00	276.42	723.58
Subsidizing work in colleges	1,600.00	......	......

INVENTORY OF RESEARCH LABORATORY

PITTSBURGH, PA.

JANUARY 1, 1921

Bank Balance at Pittsburgh		$ 141.89
Petty Cash Balance		15.35
Furniture and Fixtures		26.62
Books:		
Purchased	$ 31.04	
Presented	62.50	
		93.54
Literature, Pamphlets, etc.		19.30
Drawing Supplies on hand		45.84
Office Supplies on hand		64.05
Stationery on hand		82.24
Supplies for Heat Investigation on hand:		
Purchased	$ 202.93	
Presented	1,580.00	
		1,782.93
Supplies for Ventilation Investigation on hand:		
Purchased		
Presented		
Instruments for Ventilation Investigation on hand:		
Purchased	$ 251.80	
Presented	819.56	
		1,071.36
Instruments for Heat Investigation on hand		676.09
		$4,212.56

DISCUSSION

At the meeting of the Research Committee held yesterday a new Chairman was elected, Jay R. McColl.

J. R. McCOLL: I am very enthusiastic about the outlook. From the reports of your former Chairman and your Acting Director I feel that we have before us a very fine year and rich in results; and I shall do my best to satisfy your expectations. I have received some 25 or 30 responses to my circular letter asking for suggestions on additional subjects to undertake. Our Subjects Committee will as a result of the proposed subjects report to the Research Committee, with a view of giving Acting Director Scipio other lines to take up. There are some very good suggestions in the list and I hope the work will be enlarged very much.

FRED R. STILL: There was a matter brought up before the Council in New York on Tuesday, January 25, that ought to be reported here or acted on, in connection with the name of the Research Bureau. We have been calling it the Research Bureau, but that conflicts with the name of the Bureau of Mines and it was suggested that we henceforth speak of it and print it on our letterheads, as the Research Laboratory, to properly distinguish between the two organizations. I make a motion that this suggestion be adopted.

The motion was seconded and carried.

No. 588

THEORY OF DUST ACTION

Report Upon Investigations Carried Out Jointly by Research Bureau and U. S. Bureau of Mines

By O. W. Armspach, Pittsburgh, Pa.

Member

In reviewing the work that has been done in connection with air dust and its determination, it becomes evident that little if any attention has been given the particular action of the various kinds of dust particles when suspended in air. Practically all effort has been given to the methods of measuring the quantity of dust and its physiological effects in the body, while the action of the dust itself, when suspended in the air seems to be entirely lost sight of. A dust count made in a room gives no definite information concerning the conditions of air dustiness at the source.

In analyzing conditions of air dustiness in factories, workshops, etc., with a view of improving conditions, in addition to determining the number of particles in the air, it is imperative that a study be made of the conditions at the machines, the size of the room, the kind and size of the particles, and the relative time the particles will remain in suspension. Under certain conditions the dust content of the air will increase. Under other conditions the content may remain constant or even decrease. At the end of definite periods which may be easily calculated, the dust content will remain constant. For iron particles, the content will remain constant after the first hour; for glass, after the fourth hour; for rubber, after the seventh hour; and for leather dust approximately 15 hours are required before results in the room become constant. It is therefore essential to further progress that a theory of dust action be established. A careful study of the theory developed in the present article has compelled the following important conclusions:

1. The dust content in a room depends upon the density of the material, velocity of fall of the particles, and the size of the room; the dust will accumulate until a definite content is reached depending upon the rate of fall, and the number of air changes;

2. The total dust given off by various machines when handling different materials can be determined from the average count resulting in the room;

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1921.

3. Dust conditions can be greatly improved by providing the proper number of air changes; there is, however, a limit to this number and usually five changes per hour are sufficient;
4. For equal conditions of air dustiness, as the density of the material increases, the weight per cu. ft. decreases;
5. All dust determinations should be on a basis of number of particles per cu. ft.

Dust for our purposes may be defined as particles of matter so finely divided that they easily remain suspended in air, and the velocity of fall is comparatively low. We may consider them to be spheres. The length of time that a particle remains in suspension depends upon its diameter, its density, and the density and viscosity of the air.

Assuming no slip between the air and the particle, the velocity of fall will vary directly as the square of the radius, inversely as the viscosity of the air, and directly as the density of the material. This relation may be expressed by Stokes equation: $V = \frac{2}{9} g r^2 \left(\frac{d - d_1}{u}\right)$

in which, V = Velocity of fall in centimeters per second;
g = acceleration of gravity;
r = radius in centimeters;
d = density of the material;
d_1 = density of the air;
u = viscosity of air.

There is, however, some slip and the true velocity has been expressed by Cunningham as follows:

$$V = \frac{2}{9} g r^2 \left(\frac{d - d_1}{u}\right)\left(\frac{1 + B}{rp}\right) \quad (1)$$

B = constant;
p = barometric pressure in millimeters.

This formula has been experimentally determined by recording the time of fall of minute spheres of wax. B was found to equal 0.0075. The value of u depends upon the temperature of the air and is given by the expression: $u = \frac{150.38\ T^{3/2}}{T + 124} \times 10^{-7}$

where $T = 273.1 + t$

Assuming t to equal 21 deg. cent., or approximately 70 deg. fahr., the value of u will be 1814×10^{-7}

Substituting into equation (1) the following values: $g = 980$ cm. per second; $d_1 = 0$ (very small relative to d); $u = 1814 \times 10^{-7}$, and $B = 0.0075$ for 70 deg. fahr.; then

$$V = (1{,}200{,}540 r^2 + 11.85 r) d \quad (2)$$

By the aid of equation (2), it is possible to compute the theoretical velocity of fall of the various kinds of dust. There is, of course, an additional force due to air movement and other disturbances, but the equation has more value when comparing the relative action of the various kinds of dust.

Table 1 was computed by substituting into equation (2) 0.0001 cm. (1 micron) for the value of *r*. Column 4 gives the time required for various dusts to fall 10 ft. Cork dust for instance, will fall 10 ft. in 25.7 hours, while iron dust will fall an equal distance in 0.84 hours.

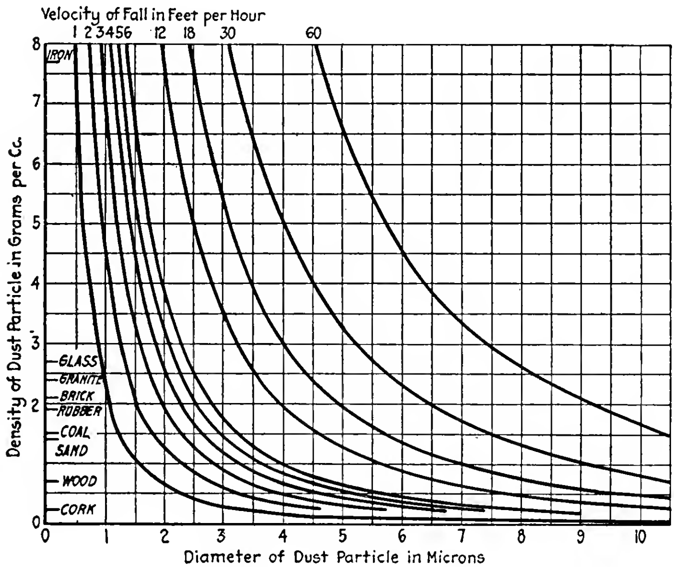

FIG. 1. CHART SHOWING VELOCITIES OF FALL FOR VARIOUS KINDS OF DUST

TABLE 1. VELOCITY OF FALL FOR VARIOUS MATERIALS 2 MICRONS IN DIAMETER

Material	Velocity cm. per sec.	Minutes per foot	Time required to fall 10 ft., Hours
Brass	0.1110	4.59	0.77
Iron	0.1015	5.02	0.84
Glass	0.0396	12.8	2.13
Granite	0.0317	16.1	2.68
Limestone	0.0356	14.3	2.38
Slate	0.0370	13.8	2.30
Sandstone	0.0304	16.7	2.78
Graphite	0.0290	17.5	2.92
Brick	0.0277	18.4	3.07
Clay	0.0254	20.0	3.34
Sand	0.0185	27.5	4.58
Coal	0.0198	25.7	4.28
Cement	0.0127	40.2	6.70
Wood	0.0093	54.7	9.15
Cork	0.0033	154.0	25.7
Ivory	0.0241	21.1	3.52
Rubber	0.0255	19.8	3.30

Fig. 1 shows that the velocity of fall of a particle is influenced both by the density of the material and the size of the particle. A particle with a diameter of 2 microns and a density of 8, will fall at the same rate as a particle having a diameter of 5.75 microns and a density of 1. The curve indicates that with a constant density, the velocity increases as the size of the particle increases, and the increase in velocity becomes greater as the diameter becomes greater—that is, for the larger particles, the friction of the air is less effective and the velocity of fall more nearly equals that due to gravity.

SIGNIFICANCE OF THE DENSITY

Fig. 1 shows that the density of the dust particle is an important factor, when the conditions of air dustiness in a room are considered. Particles of iron dust, 2 microns in diameter, will fall at a rate of 12 ft. per hour, whereas particles of wood dust of the same size would fall only 1¼ ft. per hour. Therefore, when equal quantities of wood and iron dust are produced by various processes, the conditions of air dustiness in a room will differ considerably. The average size of the particles in a steel-grinding establishment will be very much smaller than the size of the particles in a furniture factory and the means of controlling the dust as necessary to maintaining desirable conditions in the various establishments must be compatible with the fineness and the density of the material handled.

CONSIDERATION OF THE SOURCE

Since dusts of different densities fall with different velocities, the average dust count in a room will depend upon the kind of material, the conditions at the source, velocity of fall, and the length of time that the dust-producing machines have been in operation. Machines giving off equal quantities of dust but handling materials of various densities will produce widely varying conditions in a room. To illustrate, let us assume that a machine gives off sufficient dust to result in an average of n_2 particles per cu. ft. in the room.

Also let: N = the number of particles given off at the source per cu. ft.;
n_1 = number of particles per cu. ft. in the room at beginning of hour;
n_2 = particles per cu. ft. at the end of the hour;
v = the rate of fall in ft. per hour;
h = height of dust in room.

When v is less than h. When the density and size of the particles are such that the velocity of fall is numerically less than the height of the dust, we may assume that the particles liberated during an hour will not settle until after the hour. A part of the dust in the room at the beginning of the hour, however, will settle and this quantity will depend upon the height of the source and the velocity of fall. The quantity per cu. ft. at the end of the hour may then be determined as follows:

$$n_2 = n_1 - n_1 \frac{v}{h} + N \qquad (3)$$

n_1 = dust in the room at beginning of the hour per cu. ft.;

$n_1 \dfrac{v}{h}$ = that portion which settles during the hour;

N = dust added during the hour per cu. ft.

The algebraic sum of these quantities equals the dust per cu. ft. in the room at the end of the hour.

When v is greater than h. When the velocity of fall is greater than h, a part of the dust liberated any hour will fall during the hour, and the

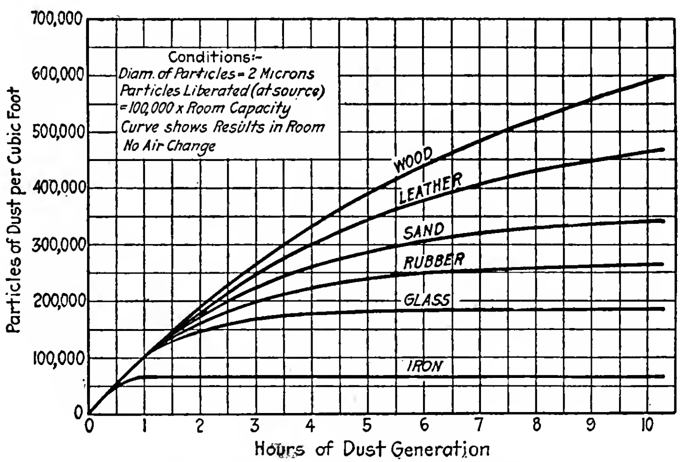

FIG. 2. CURVES SHOWING PARTICLES OF VARIOUS KINDS OF DUST PER CUBIC FOOT WITH EQUALLY UNFAVORABLE CONDITIONS AT THE SOURCE

smaller the height of the dust, the more will be the quantity that settles from the room. This is expressed by the equation:

$$n_2 = n \frac{h}{v} \tag{4}$$

To illustrate the significance of equations (3) and (4), let us assume several machines that handle various materials and liberate equal quantities of dust. In other words, the exhausting devices having the same efficiency and conditions at the various sources are equally unfavorable. For comparison, we may assume dust 2 microns in diameter. Also let N = 100,000 and h = 8 ft. The value of v for various materials is taken from Fig. 1. Substituting into equation (3) and solving for n_2 we have for glass, at the end of the first hour, 100,000 particles per cu. ft., at the end of the second hour 146,300 particles, the third hour 167,600 particles, fourth hour 177,400 particles, etc., until the sixth hour is reached when the count will remain practically constant. The results for various materials have been plotted in Fig. 2. Note that the heavier dusts reach the maximum content condition early in the day while for wood or the lighter materials, the dust content continually increases even after the tenth hour is reached. This clearly indicates the necessity of exceptional care in the design and operation of dust-preventing devices installed in establishments operating 24 hours a day.

CONSIDERATION OF THE RESULTS IN THE ROOM

A dust count in itself gives no direct information concerning the conditions at the machine causing the dust. It is necessary to go further and consider something of the size of the particles, the size of the room, and the relative time the particles of dust remain in suspension. For instance, if a dust determination indicates that the average count in both a metal-

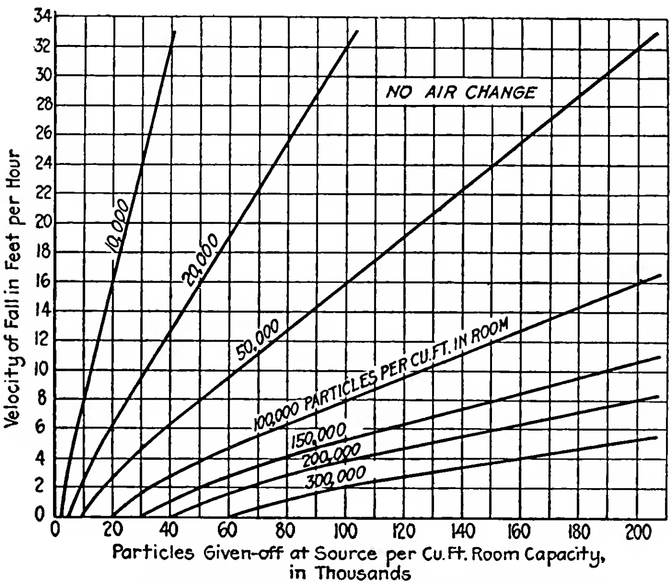

FIG. 3. CURVES SHOWING RELATION OF DUST COUNT IN ROOM TO QUANTITY GIVEN OFF AT SOURCE

grinding room and a room where wood-carving machines are in operation, is 100,000 particles per cu. ft., it does not follow that the machines are creating equal quantities of dust. This conclusion is frequently made. The exhausting devices on one machine may be much more efficient than the devices on the other machines.

Here, again, we are concerned with the density and size of the particles. We can compute by the use of equations (3) and (4), the amount of dust necessarily given off at the machines to result in a stated quantity in the room. Solving for N (the dust given off at the source), when v is less than h:

$$N = n_2 - n_1 + n_1 \frac{v}{h} \tag{5}$$

When v is greater than h:

$$N = n_2 \frac{v}{h}$$

Suppose that the dust particles are 2 microns in diameter and that at the end of the fifth hour the average count in the room is 100,000 particles per cu. ft., the machines handling different materials must necessarily give off different quantities of dust if the results in the room are to be the same in either case. Taking the value of v from the curves in Fig. 1 and substituting into equations (3) and (4), we can compute the quantity of dust resulting in the room. The results are shown in Fig. 3.

If the material is wood, then the machine is liberating 25,650 particles per cu. ft.; if the material is glass, it is liberating 55,000 particles, and

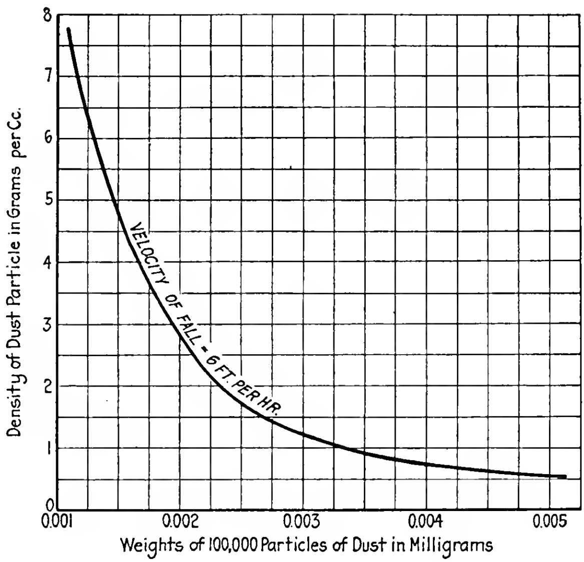

FIG. 4. CURVE SHOWING WEIGHTS OF 100,000 PARTICLES OF DUST OF VARIOUS DENSITIES FALLING WITH EQUAL VELOCITIES

if it is iron, 150,000 particles are being liberated per hour. These equations therefore, offer a means of determining the relative efficiency of different exhausting devices when handling various materials.

Fig. 3 illustrates the results at the end of the fifth hour. From the curve, an estimate of the dust liberated by the machines handling various materials may be made when the average dust count in the room is known. For example, the sample taken at the end of the fifth hour shows there are 150,000 particles of sand per cu. ft. in the room, also the velocity of fall as taken from Fig. 1 is 4 ft. per hour. Then in Fig. 3, the intersection of the 4 ft. velocity line and the 150,000 particles line would indicate that the machines are liberating 77,700 particles per cu. ft.

EFFECT OF AIR CHANGE

So far we have considered only the effects of the conditions at the source of dust and the velocity of fall of the particles. In addition, there is a decrease in the dust content due to the change of air in the room, This change may result from natural infiltration of air from outdoors or by exhausting the air from the room by mechanical means. When air leaves the room, each cu. ft. carries with it a certain number of particles, depending upon the dust content in the room. Also an equivalent amount of air must enter the room. There is, therefore, a continuous process of dilution taking place, and the dust content in the room is decreased. When the dust leaving the room due to the air change, plus the dust falling from the room, is equal to the total dust liberated, the conditions are balanced and the dust content in the room remains constant.

Suppose N particles per cu. ft. are liberated in the room per hour; also let a equal the number of changes per hour. The dust in the room

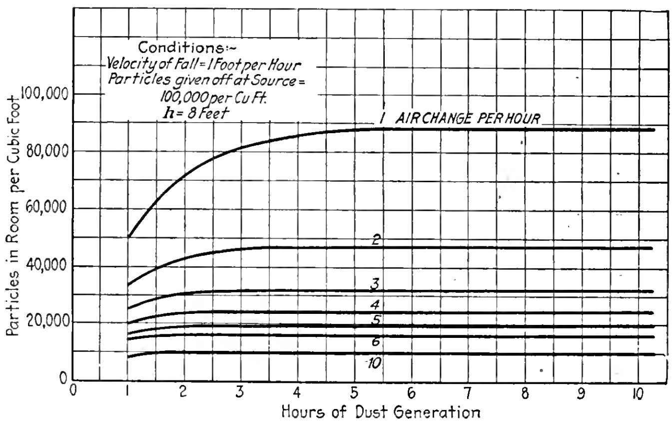

FIG. 5. CURVES SHOWING EFFECT OF AIR CHANGE ON DUST CONTENT

per cu. ft., considering only that liberated during the hour, is then represented by the equation:

$$n_e = \frac{N}{1 + a.}$$

Now suppose the room already contained n_1 particles. After the air change, of these there will remain $\frac{n_1}{1 + a}$ particles, and through falling $\left(\frac{n_1}{1+a}\right)\frac{v}{h}$ will be lost. Therefore:

$\frac{n_1}{1 + a} - \left(\frac{n_1}{1 + a}\right)\frac{v}{h}$ would represent that portion of n_1 remaining in the room, which is denoted by n_f

$$n_f = \frac{n_1}{1 + a} = \left(\frac{n_1}{1 + a}\right)\frac{v}{h}$$

Simplifying:

$$n_f = \frac{n_1\,(1 - \dfrac{v}{h})}{1 + a}$$

Hence, the total dust in the room at the end of any hour when v is less than h is

$$n_2 = n_f + n_e = \frac{N}{1 + a} + \frac{n_1\,(1 - \dfrac{v}{h})}{1 + a} \qquad (7)$$

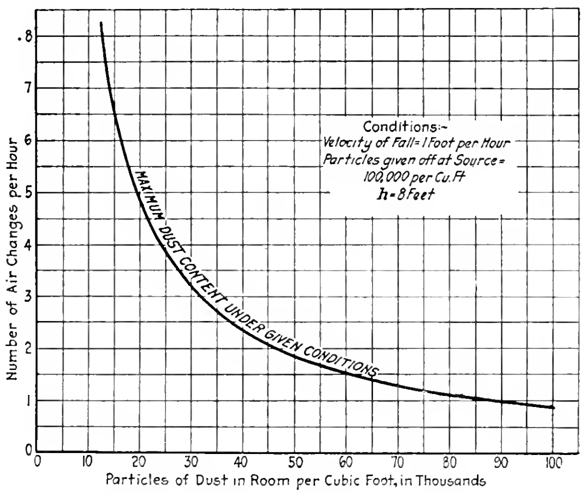

FIG. 6. CURVE SHOWING EFFECT OF AIR CHANGE ON DUST CONTENT

When v is greater than h, n_1 *is* 0, and the expression becomes:

$$n_2 = \frac{N}{a + \dfrac{v}{h}} \qquad (8)$$

The importance of providing for a definite number of air changes in dusty industries is clearly illustrated in Fig. 5. The curves have been plotted for dust 2 microns in diameter falling with a velocity of 1 ft. per minute, and 100,000 particles per cu. ft. per hour are given off at the source. Note that with one air change, the result in the room at the end of the fifth hour is 88,000 particles per cu. ft. With two air changes the result is only 47,000 particles. If no air is allowed to enter the room the result will be 389,600 particles per cu. ft., or an increase of 342 per cent over the result obtained with one air change.

Equations (7) and (8) clearly point out the improvement in the air conditions we may expect when conditions at the source are improved or when the quantity of air passed through the room is increased. It is often a difficult problem to remedy conditions in certain dusty establishments. Should more efficient exhausting devices be installed at the machines producing the dust, or would a greater improvement result by increasing the number of air changes in the room? No doubt a certain number of changes would improve conditions, but there is a limit to this number beyond which it would not be practical to go. This is shown in Fig. 6. For the given conditions the improvement obtained after the fifth air change is so small that in most cases it would not be practical to pass this point.

BASIS FOR DUST DETERMINATIONS

At present there is considerable discussion concerning instruments and methods for making air dust determinations. Some investigators advocate that determinations should be made by weight, while others recommend the counting methods. Let us look at the question from a theoretical standpoint.

Assume that, irrespective of the material, machines are giving off dust sufficient to result in an average count of 100,000 particles per cu. ft. in the room. To place the machines handling the various materials on an equal basis, it is assumed that all particles fall at the rate of 6 ft. per hour; in other words, the various kinds of particles are of such size that regardless of density, all remain suspended in the air equal lengths of time and the conditions in the room are comparable. Since the particles vary in density and since they fall with equal velocities, then the weight per unit of volume must differ for the various materials. What then is the weight of dust per cu. ft. of air for the various materials?

Fig. 4 has been constructed to show the relation of the weight of the particles to the density. The weights of 100,000 particles were computed by finding the size of the various particles necessary to result in a fall of 6 ft. per hour. For example, the weight of iron dust per cu. ft. is approximately 0.0011 mg. while the weight of wood dust for exactly the same conditions of dustiness is 0.004 mg. per cu. ft. This is an important fact. The curve shows that the lighter the material, the greater must be the weight per cu. ft. to result in equal conditions of dustiness.

Furthermore, the physiological action of the dust in the body and the extent of irritation upon the mucous membrane is a direct result of the number, irregularity, and composition of the particles, and we are, therefore, little concerned with the total weight of the material. It is quite evident then that a dust determination based upon weight is extremely misleading and must be considered unwise.

DISCUSSION

Perry West: In connection with the exhaustion of the air in these different tests, was the exhaust all taken out at the source of injection of the dust, or was some of it exhausted from other parts of the room?

The Author: What you want is air change. If the air is exhausted at the machine the dust is not getting into the room. You are removing the dust at the source. The theory will give you the proper answer in either case. You can from the dust count in the room calculate how much is being given off by the machine. Now when you apply the air change equation, I am assuming that you are supplying that air at a point different from that exhausted at the source.

Margaret Ingels: I would like to know if these figures take into account the condition of the air; for instance, that the hygroscopic material will absorb any moisture in the air more readily and become heavier; or were all the tests made under the same conditions?

The Author: I want to call your attention to the fact that this is theory. The theory will answer your question. No doubt humidity has a bearing on the length of time dust remains in suspension. If we consider the theory we will know what the effect of humidity is. Increased humidity will have two effects; one will be to change the density of the air, which if you will look at the equation, can be neglected because its effect is very small compared with that of the other factors. A change in the density of the air or the humidity will change the viscosity of the air. This factor can be neglected. However, there is another effect of humidity and that is the absorption of the moisture by the dust particle itself. It may be that this has much to do with the moisture content of the particle. From the increase in the density of the particle, you will see from the curves that this effect is not so large as you might expect. These influences are all present but they are small compared with some of the other factors. The more factors we can consider, however, the closer we are going to come to the right answer.

Willis H. Carrier: Is direct air change required? I think it was in the mind of a great many of us who have had occasion to put in dust removing systems that the air change is often very much greater than that given by Mr. Armspach in the theory. When we have machines that we can readily hood efficiently, a comparatively small air change is required; and I will say that roughly the air changes agree very closely with Mr. Armspach's figures. However, as we know, there are many installations where such an efficient hooding is impossible, sometimes where no hooding is possible, there has to be a general exhaust system. Now the amount of air required varies entirely in accordance with the dust production, I think, as he points out. Anybody that would put in six changes an hour or even ten changes an hour, would make a total failure of his installation. We know that because we have failed. So that is very good proof, and we know that air changes about double the number for the different conditions as given here are required in practice.

Now there is a reason for that. It is probably due to the inefficiency of contact, inefficiency of exhaust; the dust particles are not moved in proportion to the amount that is removed. Or on account of the necessity of removing it from the source owing to the installation requirements.

I think there is another point that Mr. Armspach spoke of and that is the electrical condition. In a good many cases, for instance, in spinning rooms of textile mills, the production of dust, and with that the lint, the fibre dust, is largely a matter of electrostatic conditions. If you have ordinary conditions without proper humidification you have very high electrostatic conditions at the point where the spinning occurs. You will see all those little fibres stick right out straight from the threads; similarly, undoubtedly, all the dust particles are being driven off at high velocity. They are repelled at high velocity and they repel each other in the air, and the dust count under such conditions probably greatly increases over what it would be in air that was not under electrostatic conditions or particles that were electrified. There undoubtedly is a great difference.

The effect of moisture, therefore, is perhaps material in removing or reducing those electrostatic conditions and producing less dust and allowing it to fall more readily. The particles that are not electrified will not repel each other, and one particle strikes another and gets heavier and falls down. Perhaps that collates more or less when it gets denser, but we find air changes in those rooms with proper humidity, as far as conditions admit it, because we find that changes higher than ten changes an hour have been required to give satisfactory dust conditions.

I think it would be well for Mr. Armspach to supplement his theory by visiting some of the plants which are typical in dust removal and see how his theory applies and where there are discrepancies to determine the reason for it.

SAMUEL C. BLOOM: Another point along the line of electrostatic element. What may be true in spinning rooms may be very different in other industries, where the electrostatic charge is of the opposite sign to what you will find on dust in spinning mills. In other words, the dust particles that are electrified to the sign opposite that of the earth and the walls of the building which are in earth's contact may act to increase the rate of precipitation and vice versa.

THORNTON LEWIS: I might point out in that connection that in my experience one of the factors in removing dust in a dusty plant that is just as important as air change is the method of contact of the air removed and the dust particles. In other words, the efficiency of your hooding where the dust is generated, provided you can hood, since there are some installations that you cannot hood. I know an installation where there are as many as 20 changes an hour, and the best hooding that was possible under the conditions, and yet the dust removal system was very inefficient. That air change reduced to 10 changes an hour produced doubly as bad results, so that the limit which Mr. Armspach suggests, or the practical limits indicated by theory, I do not believe is borne out in practice.

THE AUTHOR: I want to correct the impression that I believe you have, that five or six air changes are sufficient in all cases. From Equation 7 you can calculate how many air changes are required to maintain the dust content at a desirable point. There are certain cases in dusty industries where as high as even 20 changes would be required. The thing I want to point out is that you can calculate how many air changes are necessary for the conditions with which you are dealing, and these calculations should be made before the dust removing devices are installed. I do not want to give you the impression that six changes of air are sufficient for any condition met with.

FREDERIC F. BAHNSON: In reference to Mr. Carrier's suggestion about the electrostatic effect I would like to give you one example of an experience I had about 10 years ago. I tried to humidify the granulating room in a tobacco factory making a very well-known brand of granulated smoking tobacco, the purpose being to keep down the dust. It was almost unendurable even for the negro workmen, and the problem was how to get rid of it. You could not hood the machines, the dust was worth something for fertilizer but more if it could be kept on the leaf and sold as tobacco. The dust nuisance could not be overcome by the ordinary method of dust removal. So they decided they would like to have that room hum'dified. We stopped the dust all right, but in high humidity the tobacco was so soft it would not properly break up in the granulators.

It seems to me, therefore, based on that and a good many other experiences of less emphatic kind, that humidity does not so much accelerate the dust falling as it does prevent the dust forming; particularly so in hygroscopic substances. According to my best information you do not find between oven-dry weights and perhaps 100 per cent humidity much over 14 or 15 per cent increase in weight on wood. Unless the moisture actually condenses on the surface of the particle of wood that 10 to 14 per cent increase of weight in the particles is not going to have a very serious effect on the dust. I do believe there will be many less fine particles thrown off of wood in a highly moistened atmosphere than there are in a dry, just as we experienced in the tobacco factory years ago.

PERCY NICHOLLS: I would raise a question on the mathematics in Mr. Armspach's paper. In Fig. 2, on the simple assumptions made of treating all materials on the same basis, I see no reason why the shape of the curve for wood should differ from the shape of the curve for iron. That difference seems to me to come in due to the use of Equation 3, where the distinction is made between what happens to the particles after one hour and what happened before. The simple assumption made as far as I gather from the paper, is that any particle will fall a distance of vt in a time t; and any particle will, therefore, remain in the air for the time $\frac{h}{v}$; therefore $\frac{h}{v}$ should be the critical time. The rate of increase of dust particles for all of these various materials would be the same up to that time $\frac{h}{v}$, and thereafter would proceed in a straight horizontal line. This makes some difference in the shape of the curves, and if it is correct would change some of the curves that he has drawn. For instance, taking the curve for wood on Fig. 2, this would be a straight line at the same angle as the starting off line up to the time of about 6¾ hours; and after that time it would run along constant at 800,000 dust particles. Now drawing a curve from the origin up to 6¾ hours and 800,000 dust particles brings the point where the straight line starts outside of his figure, and, as will be seen, would make some little difference in the calculations. This will also, I believe, affect some of the other figures, especially Figs. 3 and 5, for the number of air changes, and would somewhat increase the number of air changes required for the lighter materials.

O. W. ARMSPACH: The dust content for the end of any hour is fixed by the quantity of dust in the room at the end of the previous hour, plus the dust liberated during the hour. A portion of the particles remaining in the room from the previous hour will fall from the room and for this reason the shape of the wood curve is not the same as the shape of the iron curve. The determining factor is the rate at which the particles will fall.

The calculations for the number of air changes required to maintain the dust at any desired content should be based on the maximum or constant condition and the slope of the curve up to the constant point would in no way affect the air changes required. Fig. 6 is based upon this constant maximum condition. This is shown by the following equation:

$$N = n_k \times a + n_k \frac{v}{h}; \text{ therefore } \quad n_k = \frac{N}{a + \dfrac{v}{h}}$$

where n_k = the maximum and constant dust content.

$$\text{when } a = o, \; n_k = \frac{N}{\dfrac{v}{h}}$$

In considering wood dust two microns in diameter liberated at a rate of 100,000 particles per cu. ft., then n_k = 800,000 particles as a maximum and constant condition with no air change, and this agrees with your calculation.

No. 589

EFFICIENCY OF THE PALMER APPARATUS AND THE SUGAR-TUBE METHOD FOR DETERMINING DUST IN AIR

By A. C. Fieldner[1], S. H. Katz[2] and E. S. Longfellow[3], Pittsburgh, Pa.

Non-Members

INTRODUCTION

In connection with the improvement of conditions relating to the health of workers in the mining and metallurgical industries, the Bureau of Mines has cooperated with the U. S. Public Health Service in determining the amount of injurious dust suspended in the atmosphere. The three methods for dust determination most commonly used are: (1) Washing out the dust from a measured flow of air with a water spray as in the Palmer apparatus[4]. (2) Filtering out the dust from a measured flow of air with a water-soluble granular medium as in the sugar tube method[5]. (3) Forcing a definite volume of air at high velocity through a small orifice against a coated glass plate to which the particles adhere so that they may be counted subsequently with the aid of a microscope. The Hill dust counter[6] and the Konimeter[7] depend on this principle. None of these methods are entirely satisfactory and all of them fail to include all the finer particles which are, when inhaled, the most injurious.

A critical study of the efficiency of these methods with respect to very fine particles seemed very important, especially as new means for making this study are now available in the methods developed for testing the effectiveness of gas masks and other protective devices against poisonous and irritating smokes. This paper gives the results of such a study of

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[4] Palmer, G. T., A new sampling apparatus for the determination of aerial dust: *Am. Jour. Pub. Health*, vol. 6, 1916, pp. 54-55.

[5] General Report of the Miners' Phthisis Prevention Committee: Union of South Africa, March 15, 1916, 199 pp.

Final Report of the Miners' Phthisis Prevention Committee: Union of South Africa, January 10, 1919, 110 pp.

[6] Hill, E. Vernon, Quantitative determination of air dust: *Heating and Ventilating Magazine*, vol. 14, June, 1917, pp. 23-33.

[7] Final Report of the Miners' Phthisis Prevention Committee: Union of South Africa, January 10, 1919, 110 pp.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1921. Published by permission of the Director, Bureau of Mines.

the Palmer apparatus and is the first of a series which will ultimately cover the various methods for determining dust in air.

DESCRIPTION OF THE PALMER APPARATUS

Fig. 1 shows the Palmer apparatus, which is completely built into a portable case. The washer consists of the large pear-shaped glass bulb with a "U" about 1 in. in diameter at the bottom; one side of the "U" is open to the air. Pure water to the amount of 40 cu. cm. is charged into it for a test; this amount causes the level in each arm of the "U" to rise ¼ in. above the point of separation. Air is drawn through the water by means of a small blower, operated by a direct-connected motor using power from lighting circuits. The rate of flow of the air is measured by a pitot tube and an inclined gauge, the scale being graduated in cu. ft. per minute, ranging up to 6. The rate of the flow is controlled by means of a sliding disc which closes the outlet from the blower.

To test the Palmer dust sampler, the pear-shaped glass bulb, which is the essential part, was removed from the case and connected into an apparatus for measuring its efficiency. Two methods of testing were employed. The first was based on optical comparison of the light (Tyndall effect) from suspended matter in the streams of incoming and outgoing air; tobacco smoke, and air-suspended silica dust were used as testing mediums. The second method was based on weighing the matter which passed the washer, by catching it in the small light tube of a laboratory Cottrell precipitator and determining its gain in weight.

The tests made in this manner showed that the Palmer apparatus removed not more than 13 per cent of tobacco smoke from air, measured by the method based on light reflection. By the same method of measurement, silica dust, air floated and then filtered through cotton wool so that it was very finely divided, was removed to the extent of 30 per cent when the suspension passed at the rate of 4 cu. ft. per minute,—and 20 per cent, at the rate of 3 cu. ft. per minute. On a weight basis, as determined with the Cottrell precipitator, 45 per cent of the silica dust was removed by the Palmer apparatus at flow of 4 cu. ft. per minute. On a numerical basis, the efficiency must be considerably lower than the 30 per cent efficiency found by the optical tests.

TESTS WITH TOBACCO SMOKE

Description of the apparatus for optical determinations of efficiency. Fig. 2 shows a diagram of the apparatus used for testing by the means of optical comparison, using tobacco smoke as the testing medium. Wells and Gerke[1] showed that particles of tobacco smoke measured 0.273 on the average, with an average deviation of 1.8 per cent. The tobacco smoke used by Wells and Gerke for these measurements was produced in the same way as the smoke used in the work described in this paper. Since tobacco smoke particles are of so small and uniform a size and have very little tendency to clog a filter, they make an excellent medium with which to test.

[1] Wells, P. V., and Gerke, R. H., Air oscillation method for measuring the size of ultramicroscopic particles: *Jour. Am. Chem. Soc.*, vol. 41, 1919, pp. 312-29.

At the right of Fig. 2 is a small motor-driven rotary blower which furnished air under pressure. The pressure was maintained at 100 millimeters of mercury by regulating a screw clamp on the outlet end of the apparatus. This pressure, which was sufficient to force air through the entire apparatus, against the resistance of the washer and other parts, was measured by a mercury manometer near the pump. Following the manometer, in the direction of flow, is a flowmeter which measured the rate of flow of the air. Regulation of the rate was obtained by means of

FIG. 1. THE PALMER DUST SAMPLER

the valves on the pipes near the pump. After leaving this flow meter, a portion of the air passed through the 2 in. iron pipes, arranged six in parallel, which constituted the tobacco burners.

The tobacco as used, had been molded into sticks by the following formula: Sun-dried tobacco leaf was ground and screened to the sizes needed. The sized material was mixed with potassium nitrate and rosin for binder, as well as for smoke production, in these proportions:

Tobacco, 20-40 mesh	500 grams
Tobacco, 40-60 mesh	300 "
Tobacco, 60-70 mesh	200 "
Rosin, through 100 mesh	475 "
Potassium nitrate, through 60 mesh	100 "

This mixture was pressed into a sheet-iron mold of 1 in. square cross-section and 15 in. length, open on one side and lined with paper to prevent

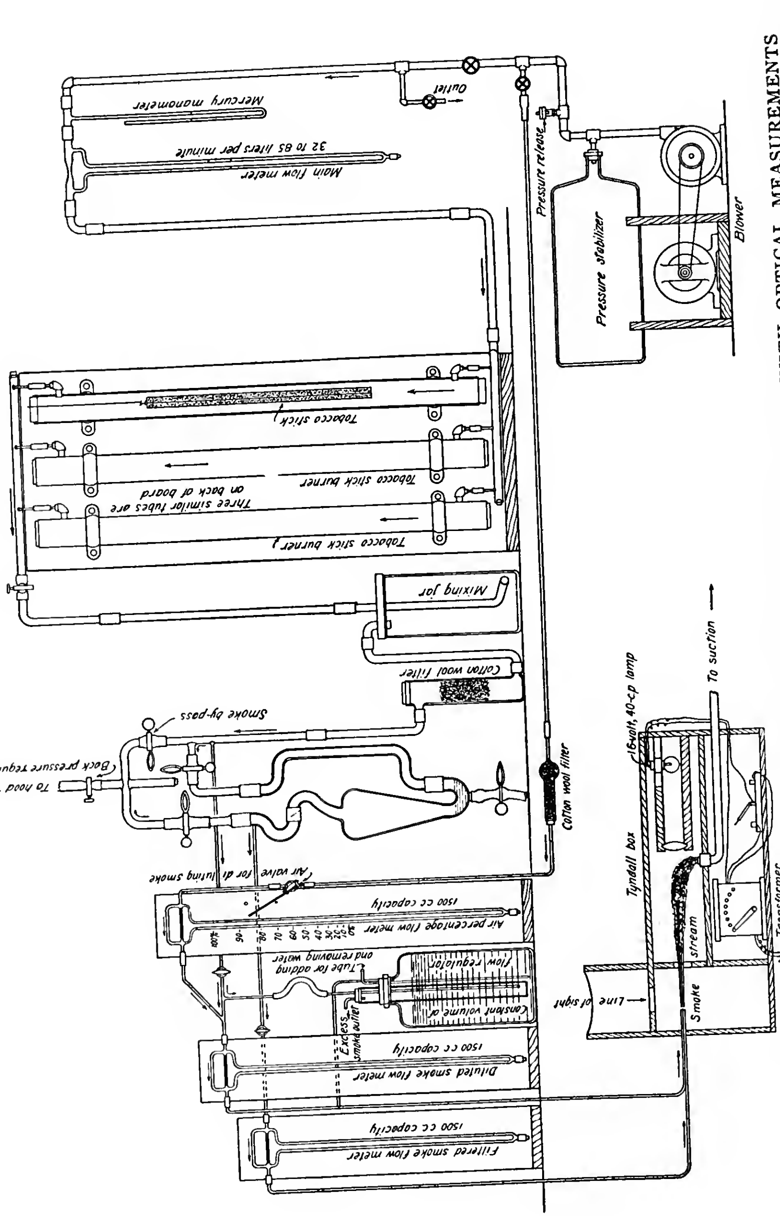

FIG. 2. APPARATUS FOR TESTING THE WASHER AGAINST TOBACCO SMOKE WITH OPTICAL MEASUREMENTS OF WASHING EFFICIENCY

sticking; candle wick was imbedded in the center of the tobacco stick to add strength. The mold and mixture were baked on a hot plate, under the cover of a half round of sheet iron, 3 in. in diameter. All four sides of the stick were baked in turn to a brown color, the mold being covered with a strip of sheet metal before inverting to bake the exposed side of the stick. Smoke from sticks prepared in this way showed no carbon and the ash was also free from carbon.

The air with smoke from the tobacco burners passed through a large jar where it was mixed with air which had been by-passed around the burning tobacco, and then through a small jar containing a wad of cotton wool for filtering out large particles. Then it passed through the pear-shaped washer of the Palmer apparatus. From the Palmer washer, the air with included smoke escaped into a hood. The tubes were also arranged so as to by-pass the smoke around the Palmer washer and to the hood directly, when desired.

The remaining apparatus was used for measuring the proportion of smoke which escaped the Palmer washer. On the outgoing side of the washer a small glass tube, connected to a stopcock and flowmeter, allowed 1500 cc. per minute of the filtered air and smoke to pass through a glass nozzle ⅛ in. in diameter, into a light beam in a Tyndall box. Smoke from the incoming side of the washer was drawn through a similar small tube, stopcock and flowmeter, and discharged through a glass nozzle in the Tyndall box, placed parallel to and ¼ in. away from the other nozzle. The two systems for conducting the incoming and outgoing smoke to the Tyndall box were made identical in size and length of tubing, constrictions and bends, so as to have equal effects of deposition of smoke particles on the walls. In order to make a comparison of the density of the two smoke streams, the incoming, as regards the washer, was diluted with pure air from the pump. This air passed through a controlling stopcock, with long lever arm for fine adjustments, and a flowmeter calibrated in percentage. It was then mixed with the incoming smoke before the smoke came to the 1500 cc. flowmeter. The latter flowmeter was equipped with the device of Oberfell and Mase[1], which automatically maintained a constant flow of 1500 cc. of gas. When no air was added, 1500 cc. of the undiluted smoke passed through the flowmeter; when 1500 cc. of air was admitted to the line, only air passed through the flowmeter and the smoke passed out of the system and escaped into the room through the regulator. The smoke left the line from a point behind that at which the pure air entered, so that no admixture of smoke and air occurred before the excess smoke escaped. When any amount of the pure air less than 1500 cc. was admitted, it passed through the flowmeter together with the amount of smoke required to make 1500 cc. and the residual smoke automatically escaped. On the flowmeter which measured the pure air, 1500 cc. per minute indicated 100 per cent. Then if the washer was 50 per cent efficient, or allowed half of the incoming smoke to pass through it, a mixture of half smoke and half, or 50 per cent of pure air, was required in the Tyndall box to match the stream of smoke from the outgoing side of the washer. After matching the smoke streams by this

[1] Oberfell, G. A., and Mase, R. P., An automatic compensating flow meter: *Jour. Ind. Eng. Chem.*, vol. 11, 1919, pp.294-6.

arrangement, the percentage efficiency of the washer was read directly from the water column of the percentage calibrated flowmeter.

The wooden Tyndall box was painted dead black inside. The beam of parallel light, opposite in direction to the streams of smoke, was provided by a lens and 40 c.p. nitrogen-filled automobile lamp. The voltage was regulated by a small transformer using 110 volt power. Suction on the box removed excess smoke. The line of sight passed through a small opening in the box, which allowed observation of about a 3 in. length of the smoke streams, just after they had passed from the nozzles.

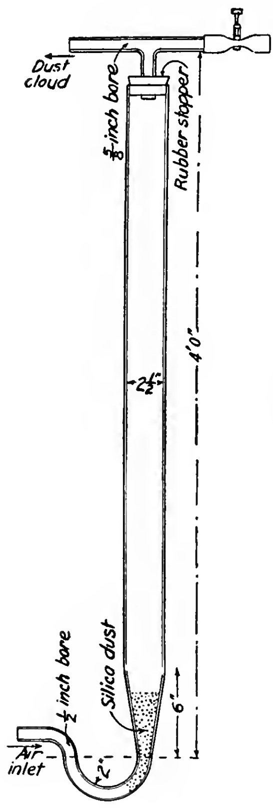

FIG. 3. APPARATUS FOR PRODUCING A SUSPENSION OF SILICA DUST IN AIR

Before measurements were made with the apparatus, it was tested blank, that is, with an unobstructed tube in the place of the washer. The smoke streams in the Tyndall box were required to match under this arrangement. In order to secure this result, the apparatus required frequent cleaning.

Procedure in Making Measurements. Experience has shown that a thin smoke served better for making optical comparisons when efficiencies were low, as in the case of the washer, so only one tobacco stick was burned at a time. When filtering efficiencies are high, denser smokes are better and six tobacco sticks may be burned when measurements of very high efficiencies are to be made. The method assumes that the actual efficiency of the apparatus being tested is independent of the concentration of the smoke. Experience in both the Bureau of Mines and Chemical Warfare Service indicates this to be true.

The accuracy of determinations decreased with the decrease in efficiency of the apparatus being tested. With the efficiencies of 98 per cent, measurements could be checked to less than 1 per cent, but when efficiencies ranged at 25 per cent or lower, errors of individual determinations might be 15 per cent. Because of the difficulty of accurate readings with low efficiencies, a number of observations were always made and, in general, this procedure was followed:

First, a reading was secured which was known to be greater than the actual efficiency by an amount just perceptible in the Tyndall stream; *then* a reading, lower in efficiency than the actual was secured in the same way. A series of such alternate observations was usually made by each of two observers. The washer gave best results when air was passed through it at a rate of 4 cu. ft. per min.; so this rate was used in the tests with tobacco smoke. At 5 cu. ft. per min. some water was carried out of the tube mechanically, while at 3 cu. ft. per min., there was less agitation of the water, less spray was produced in the tube and the washing effect was apparently poorer.

Results of the tests. Table 1 gives the results of the tests:

TABLE 1. RESULTS OF THE OPTICAL MEASUREMENTS OF THE PALMER DUST SAMPLER TESTED AGAINST TOBACCO SMOKE

Rate of flow of smoke—4 cu. ft. per min.—Water in Palmer bulb—40 cc.

		Observations		
Observers	Test No.	Perceptibly in excess of the true efficiency	Perceptibly below the true efficiency	Remarks
No. 1	1	13%		Observations even at 0 efficiency were hardly perceptibly lower than the true efficiency.
	2		2%	
	3	13%		
	4		0	
	5	12%		
	6		0	
No. 2	7	12%		
	8		0	
	9	10%		
	10		0	
	11	18%		
	12		0	
Average		13		

From these figures it is concluded that the Palmer dust sampler has an efficiency not greater than 13 per cent in removing particles of tobacco smoke from air.

Tobacco smoke consists of liquid particles. Most dusts studied for hygienic purposes consist of solid particles. For these reasons, the following tests were made with solid silica dust.

TESTS WITH SILICA DUST

Description of Apparatus for Optical Determinations of Efficiency. In order to determine the efficiency of the Palmer dust sampler in removing solid suspensions from air, tests were made similar to those with the tobacco smoke but with air-floated silica dust as the test medium. The apparatus was the same as for tobacco smoke, except for the tobacco burner. The latter was replaced by an apparatus for creating a cloud or suspension of the dust in the air. The dust-cloud producer, shown in Fig. 3, consisted of a wide glass tube, 2½ in. in diameter, tapered at the bottom over a length of 6 in. to ½ in. diameter. The narrow tube was bent into a U to retain the dust. At the top of the tube was a rubber stopper with one hole, through which passed a glass T. One arm of the T was connected with the testing apparatus; the other was provided with a rubber tube and clamp, through which dust could be allowed to escape. About ½ lb. of silica dust was heated to 200 to 300 deg. cent.

before being put into the tubes, and the air used was dried with calcium chloride. A small amount of moisture prevented formation of good suspensions.

The air carrying the silica dust was filtered through the wad of cotton wool before passing to the washer, so that only the smaller particles took part in the experiments.

With the silica-dust, precipitation occurred throughout the entire apparatus gradually, but at a rate much more rapid than that of tobacco

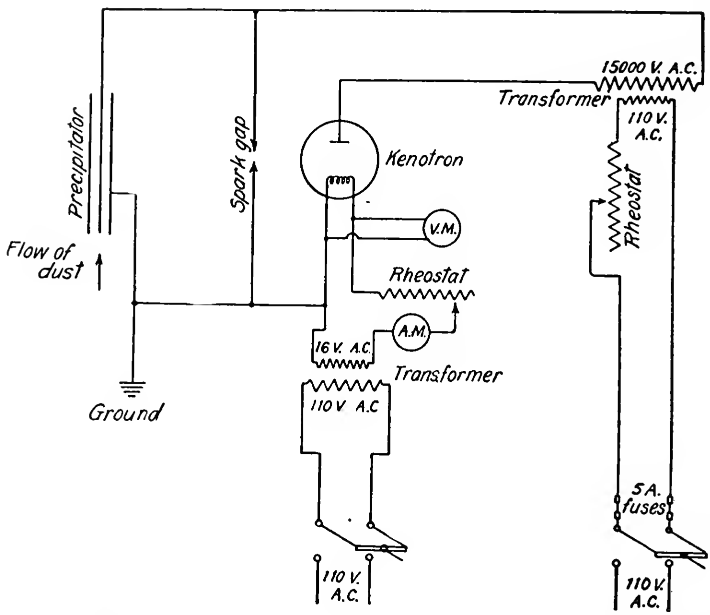

FIG. 4. DIAGRAM OF APPARATUS USED FOR TESTING EFFICIENCIES OF THE WASHER BY MEANS OF A SMALL COTTRELL PRECIPITATOR

smoke. Cleaning was required more often and greater care was used to maintain Tyndall beams of equal intensities in frequent blank tests.

Results of the Optical Determinations. The results of the tests are given in Table 2. They show that the average efficiency of the washer for very finely divided silica dust in air, based on optical measurements is only about 30 per cent. This is at least 2½ times the efficiency shown for tobacco smoke under the same conditions, which indicates a selective action upon particles of different kinds. Such differences in action may be due to differences in surface tensions resulting on contact of the different substances with water, differences in the electrical conditions of the particles, or differences in size of the particles; the latter seems most probable. Tobacco smoke particles are probably the smaller, and as will be shown later, smaller particles pass through the washer most readily.

At the rate of 3 cu. ft. per min., the efficiency against silica dust dropped to only 20 per cent. Altogether these figures show that the apparatus has an efficiency under the conditions used, which is quite low.

Description of the Apparatus and Methods for Determining Efficiency by Electrical Precipitation. The optical method of testing gave efficiency measurements based on the combined surfaces (or more exactly the light reflected from the surfaces) of the particles leaving the dust sampler, as

TABLE 2. RESULTS OF OPTICAL MEASUREMENTS OF EFFICIENCY OF THE PALMER DUST SAMPLER TESTED AGAINST SILICA DUST

Water in Palmer Apparatus—cc. 40—Rate of flow of air—Test 1—4 ft. per sec. Test 2—3 ft. per sec.

Observers	No. Readings	Efficiency Test 1 Max.	Test 1 Avg.	Test 1 Min.	Test 2 Max.	Test 2 Avg.	Test 2 Min.
No. 1	1		44%			39%	
	2		20%			18%	
	3		46%			30%	
	4		20%			12%	
	5		45%			40%	
	6		20%			10%	
Results of Readings 1 to 6		46%	32%	20%	40%	25%	10%
No. 2	7		35%			31%	
	8		23%			10%	
	9		37%			31%	
	10		14%			5%	
	11		38%			20%	
	12		18%			0%	
Results of Readings 7 to 12		38%	28%	14%	31%	16%	0%
General Average			30%			20%	

compared with the surfaces of the incoming particles. Dustiness is commonly reported in terms of mass and numbers of particles, so the following experiments were designed to give information on a weight basis. Fig. 4 shows the apparatus used. The precipitator was a duplicate of that used by Tolman, Ryerson, Brooks and Smyth[1]. Current from the laboratory service line (110 volt, 60 cycle, single phase) was passed through a transformer of 500 watt capacity which stepped it up to a maximum of 15,000 volts; the actual voltage used, probably about 10,000, was controlled by means of a rheostat in the primary circuit. This high tension current was rectified by a kenotron, which acts as a valve, thus delivering a half of the alternating cycle to the precipitator.

[1] Tolman, R. C., Ryerson, L. H., Brooks, A. F., and Smyth, H. D., An electrical precipitator for analyzing smokes: *Jour. Am. Chem. Soc.*, vol. 41, 1919, pp. 587-9.

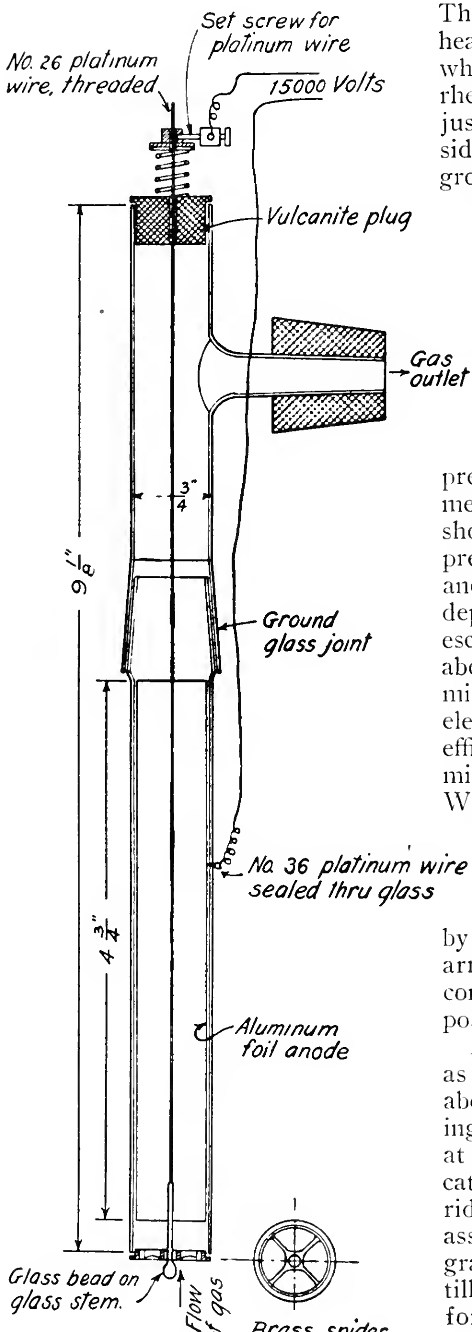

FIG. 5. DETAILS OF THE COTTRELL PRECIPITATOR TUBE

The glower coil of the kenotron was heated by a separate transformer, which delivered current at 16 volts. A rheostat in series with the glower adjusted its temperature. The anode side of the high potential circuit was grounded. Voltage across the precipitator was adjusted by stepping up to the sparking point, then reducing slightly to a brush discharge.

The Cottrell precipitator detailed in Fig. 5 was too small to care for the whole of the 4 cu. ft. of air per minute which passed through the Palmer tube, so only a portion of it, 32 liters or 1.13 cu. ft. per minute, was used in the precipitator. Tests of this arrangement by means of the optical apparatus showed that silica dust was completely precipitated at the start, but as the anode became covered by the silica deposit, an increasing amount of dust escaped. Optical measurement gave about 70 per cent efficiency after one minute for the combined washer and electrical precipitator. The average efficiency of the two during the entire minute is thus about 85 per cent. While it is possible to obtain more complete precipitation with the Cottrell precipitator by reducing the rate of gas flow, other experimental errors are multiplied by the computations required. The arrangement described was therefore considered satisfactory for the purpose in view.

Aluminum foil tubes, which served as anodes of the precipitator, weighed about 0.5 grams each. Before weighings, the tubes were dried in an oven at 100 deg. cent. and cooled in a desiccator containing fused calcium chloride. Weighings were made on an assay balance, sensitive to 0.01 milligram. An amount of 40 cc. of distilled water was used in the washer for each test. After the 4 cu. ft. of air with suspended dust had been passed through the apparatus, the water with the dust caught was carefully rinsed into a beaker and then filtered through a 9 cm. analytical filter paper of close texture (C. S. and S., No. 589, blue ribbon) designed for retaining the finest precipitates.

The silica particles were so fine, however, that the filtrate showed considerable turbidity which persisted even after passing through several separate papers or double papers. Examination under the microscope by Reinhardt Thiessen of the Bureau of Mines showed that most of the particles in the filtrate were of the order of 0.25 microns in diameter or smaller, and the largest were 2 microns in diameter. To obtain an estimate of the weight of dust in the filtrate, turbidity standards in water were made with weighed quantities of the finest air-floated dust, the range covered 10 to 1000 mg. of dust per liter of water. Comparison of the filtrates and standards in similar containers and measurement of the volumes of filtrates gave means for calculating the weight of dust in the filtrates. The filter papers with the dust caught by them were ignited and the dust residue weighed. This weight added to that determined by turbidity, gave the weight of dust caught by the washer. The sum of the weights of dust caught by the washer and Cottrell precipitator is actually somewhat less because even the two collectors had a combined efficiency less than 100 per cent by optical determination. However, the result obtained gives a fairly close approximation.

TABLE 3. RESULTS OF MEASUREMENTS OF EFFICIENCY OF THE PALMER DUST SAMPLER ON A WEIGHT BASIS, DETERMINED WITH SILICA DUST CLOUDS AND A COTTRELL PRECIPITATOR

Time of Sampling, Seconds	60	60	60
Data from Palmer Washer:			
Water used, cc	40	40	40
Air passed, cu. ft	4	4	4
Weight of dust recovered, mg	35	32	22
Weight of dust in filtrate by turbidity, mg	4	3	3
Total dust caught by Palmer tube, mg	39	35	25
Data from Cottrell Precipitator:			
Air passed, cu. ft	1.13	1.13	1.13
Weight of dust caught, mg	10.1	13.8	10.7
Final Efficiency measured optically of the combined Palmer and Cottrell apparatus, per cent	70	70	67
Total Dust which passed through the Palmer tube, mg.	35.7	48.8	37.9
Total Dust carried by the air sampled, mg	74.7	83.8	62.9
Efficiency of Palmer Dust Sampler on a weight basis, per cent	52	42	40

Results on a Weight Basis. Table 3 gives the results of the efficiency determinations on a weight basis. In the last column, are the limiting maximum efficiencies, 52, 42, and 40 per cent, found by three determinations. The average is 45 per cent. Actual efficiencies are believed to be not much less than the figures determined because the dust which escaped was probably so impalpably fine that its weight was of little consequence, although on the basis of surface or numbers the loss was probably large. The efficiency determined by weight averages 50 per cent greater than that determined optically.

The results of the weight determinations show even a lesser efficiency than that found by Bill[1], who reports 63.0 per cent efficiency on a weight basis for the Palmer dust sampler as compared with an electrical precipitator, as an average of 71 determinations made on different factory dusts. The efficiencies determined by Bill and by the authors may be considered of the same order, and the differences are probably due to differences in the character and size of the particles handled.

DISCUSSION OF THE RESULTS

The optical methods of testing gave efficiencies based on the surfaces of the incoming and outgoing particles. Dustiness is commonly reported in terms of mass and numbers of particles, so it will be well to inquire into these relations as they concern the efficiencies determined by the two methods employed.

The aggregate surface of a given mass of silica particles uniform in shape and composition increases as the first power of the decrease in diameter of the particles. The numbers of particles in a given mass of dust, uniform in shape and composition, varies inversely as the third power of the diameter of the particles. Thus, both surface and numbers of particles of a given aggregate mass increase with the fineness of division; but the number of particles increases at a rate very much greater than the surface.

On the basis of the above statements it may be that a small number of large dust particles has a greater mass but smaller surface than a large number of small particles. If the Palmer apparatus were more effective in removing large than small particles from the air, it would thus be possible with dust of mixed sizes for it to show lower efficiencies optically than by weighing. Since the optical efficiencies were lower, it is concluded that the apparatus is more efficient for determining dust of large-sized particles.

Since the numbers of dust particles increase at a very much faster rate with fineness of division than the surface, and lower optical efficiency was found, it is concluded that the efficiency of the washer on a numerical basis must be even lower than that determined optically.

Tobacco smoke, according to Wells and Gerke, consists of particles of quite uniform diameter, hence they may be taken as uniform in surface and weight. The efficiency of the washer, less than 13 per cent, found by the optical method, is then a measure of the weight and number efficiency.

SUMMARY AND CONCLUSIONS

The efficiency of the Palmer dust sampler was determined by the use of tobacco smoke and of silica duct. Two methods of testing were used. *The first* determined efficiency on the basis of the surface of the particles entering the apparatus as compared with the surface leaving, by the use of the Tyndall effect. *The second* method, based on the weight of dust,

[1] Bill, J. Penteado, The electrical method of dust collection as applied to the sanitary analysis of air: *Jour. Ind. Hygiene*, vol. 1, 1919, pp. 323-42.

used a small laboratory Cottrell precipitator in series with the Palmer washer.

The tests showed the following:

1. The Palmer dust sampler retains about 45 per cent by weight of air-floated silica dust when air is passed through it at the rate of 4 cu. ft. per minute;
2. The surface efficiency, with silica dust, measured by the Tyndall effect was 30 per cent when the air passed at a rate of 4 cu. ft. per minute, and 20 per cent at 3 cu. ft. per minute; at 5 cu. ft. per minute, water was carried mechanically from the washer;
3. The efficiency based on numbers of the silica dust particles is probably lower than the surface efficiency;
4. The Palmer apparatus was less than 13 per cent efficient in retaining tobacco smoke, as measured by the Tyndall effect.

   Assuming tobacco smoke particles to be of uniform size, the optical efficiency of less than 13 per cent should also apply to weight and number efficiency.

THE SUGAR-TUBE METHOD

INTRODUCTION

IN connection with investigations relating to the improvement of health conditions in the mineral industries, the Bureau of Mines has made considerable use of the Sugar-Tube Method for determining the nature and amount of silicious dust in suspension in air. The method now consists in passing by means of a calibrated foot pump, a measured volume of the dusty air through a tube containing granulated sugar which filters out the suspended dust. The sugar tube is then sent to the laboratory, the sugar is dissolved in water, and the insoluble dust filtered off, ignited and weighed. A portion of the sugar solution containing the dust in suspension is also made up to a measured volume, and an aliquot part (1 cubic millimeter) is placed under a microscope for counting the number of particles and measuring their size, shape and other characteristics. The method is largely adopted from those developed and described by the Miners' Phthisis Prevention Committee of the Union of South Africa[1], especially as regards counting of particles. The Bureau's contribution is principally a study of the factors which affect the filters and subsequent process of determination.

This paper briefly describes the sugar tubes and some accessory apparatus now used by the Bureau for taking dust samples. It gives the conclusions from measurements of the efficiencies of the sugar tube by methods which have been previously described.[2]

[1] General report of the Miners' Phthisis Prevention Committee, Union of South Africa, March 15, 1916, 199 pp. Final report of the Miners' Phthisis Prevention Committee, Union of South Africa, January 10, 1919, 110 pp.

[2] Katz, S. H., Longfellow, E. S., and Fieldner, A. C., Efficiency of the Palmer Apparatus for Determining Dust in Air, Jour. Ind. Hygiene, vol. 2, Sept. 1920, pp. 167-81. *Jour. Am. Soc. Heat. & Vent. Eng.*, vol. 26, Nov. 1920, pp. 687-700.

The first method was based upon comparisons of the Tyndall effects of incoming and outgoing air as regards the sugar tube. Silica dust and tobacco smoke were used as mediums for testing. The Tyndall effect gives efficiencies depending upon light reflected from the aggregate surfaces of the dust particles. The second method gave efficiencies on a weight basis; dust was evolved from a small weighable cloud producer and passed through a sugar tube. The weight efficiency was found by determining the loss from the cloud producer and comparing this with the weight of dust recovered from the sugar tube.

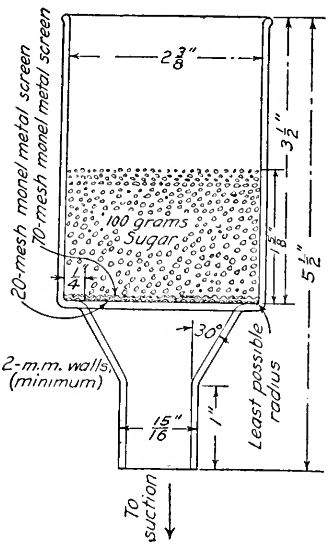

FIG. 6. A RECENT DESIGN OF SUGAR TUBE

As a result of the test the sugar tube now in use has been designed. It gives higher filtering efficiency combined with low resistance to passage of air.

DESCRIPTION OF THE APPARATUS NOW USED

Fig. 6 shows a recent design of sugar tube. It consists of a tube 2⅜ in. in diameter by 3½ in. long, having a restriction perpendicular to the wall at one end to support the screens which hold the sugar, and tapering thence to a smaller tube for attaching to the rubber hose leading to suction. The tubes have been made of glass principally because glass does not add particles to the sugar either by corrosion or erosion, and because of its low cost. For transportation the tubes are closed at both ends with stoppers of either rubber or cork composition and packed in mailing tubes. The monel metal screens are held in position by friction. This is not altogether satisfactory because the screens may become displaced in the mails. Another tube has been used which has a slight bulge for retaining the screens at the restriction, but breakage of these tubes in the mails has been found greater. Brass tubes have been used for laboratory experiments. They are unbreakable and screens may be soldered in place. Brass cannot be used for samples which must be shipped, because of corrosion on continued exposure to damp sugar. Monel metal may be used, but its adoption has been prevented because it costs and weighs more than glass.

Granulated sugar of special purity such as high-grade package sugar, is used. Most of this sugar passes a screen of 14 meshes to the inch and lodges on one of 65 meshes to the inch. One hundred grams put into a tube described above and made compact by tapping gives a layer 1⅝ in. deep. For special purposes in the laboratory, pulverized sugar of 48 to 150 mesh size was used. The latter has a higher filtering efficiency, but a higher resistance to flow of air through it and increases the work of sampling. It cannot be purchased in size so was prepared by screening XXX pulverized sugar; 60 per cent is 48 to 150 mesh size.

Fig. 2 pictures a sugar tube attached by a non-collapsible rubber hose to the foot pump for drawing the air. The pump is a "lungmotor" designed for resuscitation purposes but modified by changing the valves so that it draws air through the sugar on the upstroke and discharges on the downstroke. It is constructed of nickeled brass and aluminum, and weighs 11 lb. About 20 to 22 strokes are required per cu. ft. of air; calibration is accurately done with the entire sampling apparatus connected to a gas meter at the intake, as pictured in Fig. 7. The manometer shown was used in experiments for indicating suction. A counter is attached to the pump for registering the strokes. Approximately 15 cu. ft.

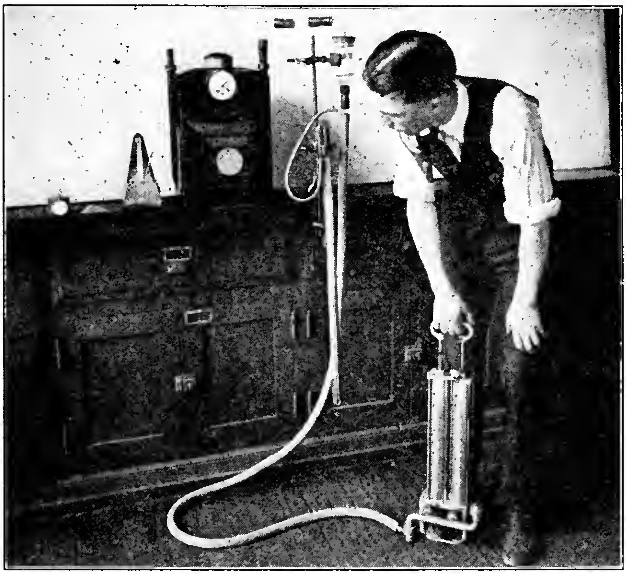

FIG. 7. SUGAR TUBE AND FOOT PUMP ATTACHED TO GAS METER FOR CALIBRATING

of air is used for each sample requiring about 15 minutes pumping. This is laborious but after some practice a sampler does not become overtired by it. A motor-driven pump has been designed for use where electric power is available.

CONCLUSIONS FROM TESTS OF SUGAR TUBE FILTERING EFFICIENCIES

A detailed description of the tests made to determine the efficiency of sugar tubes as filters will be published later by the Bureau of Mines. Some of the conclusions drawn from these tests are as follows:

1. Filtering efficiency of sugar tubes increases with increase in diameter of the tubes. Various tests made with tobacco smoke to compare tubes of 1½ and 2⅛ in. diameter showed efficiencies between 23 and 60 per cent with an average difference of 13 per cent in favor of the wider tube.

2. The rate of flow of air through unit cross sections of sugar tubes is smaller as the tube increases in diameter. Hence from number 1, it may be stated that the efficiencies of the filters decrease as the rate of air passage increases.
3. The effect of wetting sugar with water and with 35 per cent alcohol and testing by the Tyndall method against silica dust showed an average increase of 4 per cent for the water and 5 per cent for the 35 per cent alcohol over comparative efficiencies of dry sugar. Averages for similar tests showed a minimum efficiency of 57 per cent with dry filters after one-half minute of use and a maximum of 94 per cent for wet sugar after three minutes of use. Increase of efficiency with use was due to clogging by the dense dust clouds in the tests.
4. Sugar of different sizes tested with tobacco smoke showed an increase in filtering efficiency with increasing fineness. In a tube 2⅛ in. in diameter, the efficiency ranged from 27 per cent for sugar 20 to 35 mesh, to 60 per cent for 48 to 65 mesh. Pulverized sugar of 48 to 150 mesh sizes gave over 90 per cent efficiency.
5. Tests with tobacco smoke were made to show the effects of increasing the depth of granulated sugar. They showed 14 per cent for 1 in. depth, and 38 per cent for 4 in. depth. The effect of increasing the depth grew less as the height increased.
6. Tests by the method of weighing silica dust before and after producing clouds and passing them through filters, then recovering and weighing the dust caught by the sugar, showed efficiencies of dust recovery ranging from 70 per cent up to 95 per cent.
7. A sugar tube 2⅜ in. in diameter containing 65 grams of pulverized sugar, 48 to 150 mesh size, was found to give over 92 per cent efficiency against wet tobacco smoke for 15 minutes. This is the highest efficiency found in a practicable sugar tube.

Acknowledgment is due to the Miners' Phthisis Prevention Committee of the Union of South Africa, who introduced the foot pump and sugar-tube method for determination of dust in mine air; to Daniel Harrington, supervising mining engineer, Bureau of Mines, in charge of dust determinations in metal mines, and to W. A. Selvig and G. E. McElroy, who assisted in designing the present field equipment. The authors are also indebted to members of the Research Bureau of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for many valuable suggestions on their dust problems.

DISCUSSION

J. R. McCOLL: Does not the discharging of the sugar split up some particles of the dust into more particles and make the count larger?

THE AUTHOR: That is very possible in certain types of dust. In the work that has been done in the Bureau of Mines we have used tobacco smoke and silica dust. In using the silica dust, which is the thing that may be called a practical test, we have found no such noticeable effect. The particles are apparently individual and they have very little tendency to conglomerate.

No. 590

PHYSIOLOGICAL HEAT REGULATION AND THE PROBLEM OF HUMIDITY

By E. P. Lyon[1], Minneapolis, Minn.
Non-Member

THE human body is a thermostat. The temperature of the body, that is, the internal parts, is constant. By constant is meant exactly what engineers mean when they say that the temperature of a room with thermostatic control is constant. It really varies somewhat, and the small variations are made the basis of regulation.

The ancient philosophers and physicians could not understand how the warm blooded animals maintained their temperature. They looked upon animal heat as mysterious and in the category of spirits, like magnetism and electricity. We know now that animal heat is the same as other heat, and that the body obeys all the laws of thermo-dynamics, including the fundamental one of conservation of energy.

The body is a machine for transforming energy. One aspect of this transformation is the developing and regulating of heat. The constant temperature of the body is an expression of the fact that the body loses its heat as fast as it produces it, or produces heat as fast as it loses it.

In the bacteriological laboratory at the University of Minnesota are two incubators, automatically controlled. One is so arranged that it turns on an electric stove whenever the temperature falls below a certain point, and shuts it off again when the temperature rises above that point. The body uses this principle to some extent.

The other incubator (a cool one) has a net work of water pipes. When the temperature rises the city water is turned on and circulates through these pipes and carries off the heat. When the temperature falls the water circulation is shut off. If we cooled and recirculated this water, as is done in an automobile and as the body does the blood, we should have an exact example of the other principle of heat regulation used in the body.

When the automatic control fails to adjust one of these incubators to the temperature for which it is "set," the temperature of the incubator may go too high or fall too low. In either case the bacterial cultures kept in the incubator may die. This is exactly true for the human body. The body is an incubator for the millions of living units (cells) of which it is composed. If the temperature goes too high, we call the condition "fever," and the cells suffer. If the temperature goes too low, we speak

[1] Professor of Physiology and Dean of the Medical School of The University of Minnesota. Paper presented originally before Minnesota Chapter; also at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1921.

of "subnormal temperature"; and this likewise is injurious and sometimes fatal to the cells and therefore to the person whose body is composed of these cells.

Such failure of regulation, in the case of the incubators, might be due to disorder (disease) within the apparatus, or it might be due to such unusual outside conditions that adjustment would be impossible. Both contingencies find absolute counterparts in the operation of the regulatory mechanism of the human body.

HEAT PRODUCTION AND LOSS

The body produces heat by oxidizing foods. The steps in the process are complicated, but in the long run, the result is the same as if the food were burned in a furnace. One gram of starch or of fat will give in the body exactly as many calories as if burned in a bomb calorimeter, and the waste products are the same.

The chief place of heat production is the muscles. When muscles act, that is, contract, they produce much more heat than when they are resting; but resting muscles produce some heat all the time. Probably all living matter produces heat in certain phases of its chemical activity. Moreover some muscles, for example the heart and the respiratory muscles, are acting all the time. Thus the body is constantly producing heat. Unless it loses heat as fast as it is produced, the temperature must rise, and the opposite is equally true.

Since the heat production may vary, one can readily understand how body temperature may be regulated from the production side. Partially we do this voluntarily. On hot, humid days we avoid muscular exertion as much as possible; that is, we produce as little heat as possible. In cold weather we exercise strongly and produce more heat.

But partially the regulation of production is unconscious and automatic (involuntary). Shivering is a visible example of involuntary muscle contraction to produce heat, but there is another type of heat production in muscle. In hot, moist, atmospheric conditions, the muscles are partially relaxed and produce little heat; in cold weather they are tightened up and produce more heat. Physiologists call such tightening "muscle tonus." It is controlled through the nervous system by what we call reflex action, which will be referred to later."

The more important element of body temperature regulation is, however, on the heat loss side. This corresponds to the second type of incubator previously mentioned.

The body loses heat in several ways. Usually it is warmer than the substances (air, especially) in contact with it. Therefore according to the laws of physics, heat will be conducted, as heat, into neighboring matter. The rate of such transmission varies with the difference in temperature of the substances in contact; say, body surface and air. This difference in temperature may vary from the outside in a variety of ways. By use of a fan we can renew the layer of air frequently and thus increase temperature difference and heat loss. By clothing we can

[1] Some think the increased oxidation produced in the muscles by cold is indirectly produced through the thyroid glands. But the effect on heat production is the same in either case.

keep the layer of air in contact, raise its temperature and diminish the rate of heat loss by conduction, etc.

Part of the surface of our body is turned in to form the lungs and alimentary canal. As long as the temperature of the air breathed is below the body temperature, the air will be warmed in the air passages and lungs by conduction. Similarly, food or drink of a temperature less than that of the body will be warmed by conduction in the alimentary canal.

The rate of heat loss by conduction depends on the heat capacity and conductivity of the material in contact with the skin. For example, a marble floor under one foot feels colder than a rug under the other foot, though the temperature of both floor and rug are the same. We say the marble is colder. But really it is not the marble but the skin of the foot placed on it that is colder. The marble conducts off the heat faster than the rug and can hold more of it. The foot telegraphs to the brain that the foot is cold. The brain jumps at conclusions, as it often does, and says the marble is cold.

This same principle must be extended to moist and dry air. Moist air has a greater heat capacity than dry air of the same temperature. Hence at temperatures around the freezing point and somewhat above, moist air "feels" colder than dry air of same temperature. The moist air takes away heat from the skin faster; therefore, the skin is cooler. The brain which was created before the thermometer and has never gotten over its primitive errors of thought, "says" the damp air is "cold." This is particularly true when the water of the air is in the form of droplets; e. g., fog.

Conditions are very different at higher temperatures, where atmospheric humidity, by hindering evaporation, is a heat preserving factor for the body. Thus arises the anomaly that humidity may keep us warm or cool, depending on the temperature of the air in which the moisture is held.

If the temperature of the material in contact with either the outside or inside surfaces is the same as that of the body, no heat can be lost by conduction.

Much the same may be stated in regard to the second method of heat loss; namely, radiation. By this is meant loss of energy in the form of ether waves. This form of heat loss occurs only from the skin surfaces. No heat can be lost by radiation, (e. g. in a room) if all the surrounding objects and walls are of the temperature of the body.

The third way in which the body loses heat is by evaporating water. This occurs at both the skin surface and the lung surface. The rate of evaporation and consequently of heat loss, varies with numerous factors. It increases with temperature of the air on the skin, and therefore acts, in general, opposite to conduction and radiation, which is a fortunate fact. Evaporation increases with increased renewal of air, consequently a fan or wind cools the body; as likewise does rapid breathing, e. g., the panting of a dog.

Evaporation rises with increase of the surface of liquid exposed. We expose more liquid surface when we sweat than when the skin is, as we say, "dry." Of course it is never absolutely dry, and insensible perspiration is constantly being evaporated. Clothing tends to keep a layer of saturated air between the skin surface and the general atmospheric environment and consequently, always restricts heat loss by evaporation, just as it does by conduction and radiation. Naturally, different types of fabric vary in conductivity, porosity and other qualities.

In accord with the laws of physics, the body cannot lose heat by evaporation in air already saturated and of body temperature. That is the reason, the so-called "steam room" of a Turkish bath feels so hot; it is not above body temperature, but on the contrary lower. It is not the *room* that is hot; it is the *skin*.

Again if these principles are kept in mind it will be understood why the "dry" room at 165 or even 180 deg. fahr., "feels" cooler than the "steam" room at 90 deg. fahr. The skin is actually cooler in the former, because it is able to lose heat rapidly by evaporation, although it may be gaining some heat by conduction and radiation at the same time.

By the three methods, namely, conduction, radiation and evaporation, the body loses all the heat that it makes. All these vary in rate with external and internal conditions. All of them become ineffective under conditions of high temperature and high humidity. Hence, the deaths in the "black hole of Calcutta," and the so-called "sunstrokes" on a hot, humid day.

METHODS OF INTERNAL ADJUSTMENT

Various external factors have been referred to that affect the rate of heat loss by the three methods of conduction, radiation and evaporation. The body from its side is not passive. It has two ways of accommodating or adjusting itself to the three methods of heat loss and thus keeping a constant temperature. The first is by the distribution of the blood. When the outside temperature is hot, much blood is sent from the internal parts to the skin. The skin temperature is raised by this warm blood and the rate of heat loss by conduction and radiation is increased. When the outside temperature is low, little blood is sent to the skin and less heat is lost by conduction and radiation.

The second method of internal adjustment is the sweat secretion. This liquid which is practically nothing but water is extracted from the blood by coiled tubes, called sweat glands, located in the skin. When the temperature is high, the amount of secretion increases; the liquid surface exposed to the air increases; evaporation increases; the heat loss increases, and the temperature falls.

From the physical side, these processes are easily understood. From the internal or physiological side, their understanding requires that something be known of the properties of the blood vessels and something concerning the nervous system. The arteries of our body are not like the rigid pipes used for distributing water and steam. For the present purpose, it must be understood that the walls of the arteries contain rings of muscle. If the muscle rings contract, the tube becomes smaller; if the muscle relaxes, the tube becomes larger. By the word "contract" is meant shorten through living properties, like the biceps or other skeletal muscles. It does not mean the elastic "contracting" of a spring or rubber on release of stretching force.

It will now be perfectly clear how the distribution of blood is effected for purposes of heat regulation. If the body is producing too much heat, the blood vessels of the skin expand or "dilate"; those of the interior, on the other hand, contract or "constrict." The heart pumps the blood everywhere, but naturally more blood goes to the skin; more heat is lost; the temperature tends to fall. On the other hand, if the body is losing heat too fast, as when the outside temperature is low, or there is a "draft," or it has not sufficient clothing, the outer blood vessels constrict; the inner ones dilate, and the blood goes in greater abundance to the internal organs and less to the skin; less heat is lost, and the temperature of the body is preserved.

It is a beautiful machine, this so-called vaso-motor or vessel-moving mechanism. Heating and ventilating engineers, with all their control of modern materials and modern knowledge of the forces of nature, could not invent a better one.

But, let us go one step further and ask how it is that the muscle layers of the arteries all over the body can act together in such harmony, so as to increase or decrease the supply of blood to the skin in correspondence with the rising or falling temperature of the body. This is where the nervous system comes into play. We call this method of nerve control "reflex action."

REFLEX ACTION

Reflex actions are so important a feature of heat regulation that what is meant by that phrase, or its shortened form, "reflex," will be explained. It is, of course, well known that one's central nervous system consists of the brain and spinal cord. While these are convenient descriptive divisions, they constitute really one continuous mass of nervous material. The word "central" is doubly significant, for in addition to the idea of importance which the lay person would attach to the expression "central nervous system," it carries exactly the meaning which a telephone engineer would attach to the word "central." Except for its subtle and unknown connection with thinking, the central nervous system is strictly an enormously-complex automatic switchboard, with millions of connections. Indeed, while we cannot conceive how the switchboard can think, we get more than a glimpse, as we study this marvelous mechanism, both as to how the materials (sensations) on which thought is founded are secured and manipulated; and how thought can project itself into action. Both are accomplished through this beautiful automatic switchboard, the central nervous system.

The nervous system is connected to nearly all parts of the body by conductors called nerves. Indeed, it is better to speak of the nerves as part of the nervous system, just as we would speak of the wires and cables as part of a telephone system. A nerve is not a single conductor but is really a cable or bundle of separate insulated conductors or "nerve fibers." Each particular nerve fiber forms a conductor from the nervous "central," e. g., brain or spinal cord, to the particular distant organ with which that fiber is associated. Again, we see the analogy to the telephone system. But here one point of difference must be noted.

Part of the nerve fibers are strictly "afferent," that is, they conduct toward the "central." Others are strictly "efferent," conducting out from the "central." A given conductor or nerve fiber, is never used in the body for messages in both directions, as is the case with telephone or telegraph wires.

One more fundamental fact:—just as all electric currents are essentially alike and, just as the difference in results obtained from commercial electric currents, such as heat, light and motion, depend on the apparatus to which the current arrives, so in the body all nerve currents (we call them "nerve impulses") are alike, and the differences in results are due not to different kinds of impulses, but to the different kinds of organs that receive the impulses.

The "central" nervous system, or brain and spinal cord, switches these impulses in accordance with definite pathways laid down in the system and in accordance with what pathways are open under different bodily conditions. These switchings of nerve currents and the resulting responses, when involuntary, are called reflex actions. For example, if one should breathe in some dust, the particles would irritate or stimulate nerve endings in the nose; nerve currents thus started would travel over nerve fibers to a certain part of the brain; from there they would be switched over or "reflected" to nerve fibers leading out to the muscles of the chest; these muscles would contract sharply; air would be violently driven out through the nose and the result is called a sneeze. It is a reflex or involuntary act.

BODY THERMOSTAT UNDER REFLEX CONTROL

The thermostatic control of the body is effected by widespread reflexes. Let us suppose one steps from a cold into a warm room. The heat stimulates certain nerve endings in the skin; nerve currents or impulses are aroused; these currents travel into the brain and spinal cord; they are switched or relayed to other nerve fibers leading out to the blood vessels. The muscle of the blood vessels of the internal organs is stimulated and contracts, so that the vessels are smaller; therefore, less blood goes there. At the same time, the blood vessels of the skin dilate, and more blood goes there. Thus heat is lost, and the body accommodates itself to the warmer room.

If the room is very warm, especially if it is also humid, heat may not be lost fast enough by radiation and conduction and the temperature of the skin may rise. Presently through the stronger stimulation of the same nerve endings or stimulation of others not previously in action (we do not know as yet which of these is true), nerve currents are switched along the fibers that lead to the sweat glands, and evaporation comes to the assistance of conduction and radiation.

If a person is working hard his body sweats. In this case the reflex is the same, the warming of the skin being due to the flow of hot blood from within.

One can easily construct the explanation, *mutatis mutandis,* for other external or internal conditions.

Table 1 shows the percentage of heat loss by the various methods, as determined by physiologists.

TABLE 1. TOTAL HEAT LOSS FROM THE HUMAN BODY

	Vierordt		Atwater Resting Man		Atwater Working Man	
	Cal.	Per cent	Cal.	Per cent	Cal.	Per cent
Urine and feces........	47.5	1.8	31	1.4	26	0.6
Warming Air	84.5	3.5				
Evapor. from lungs....	182	7.2	548	24.2	859	20.3
Evapor. from skin.....	364	14.5				
Radia. and conduct.	1792	73	1683	74.4	3440	79.1
	2470	100%	2262	100%	4225*	100%

* Also did external work equivalent to 450 calories.

From the foregoing it will be clear that these figures represent particular cases. The relative losses may vary greatly from these figures. Everyone can see how numerous are the factors both external and internal, to which the heat regulating devices of the body are able to adjust. Of course this means that we are adapted to live under a great variety of conditions. In other words, the range of the normal is wide; and it is unscientific to say this or that is the best temperature, this or that is the best humidity, this or that is the best air movement.

PRACTICAL IMPORTANCE OF HUMIDITY

Man adjusts himself to every degree of atmospheric moisture from the practically absolute dryness of sub-zero air to the saturated air of tropical forests. Humidity is of little importance except when considered in connection with other conditions, particularly temperature and air movement.

The most usual condition under which the body heat control breaks down is high humidity and high temperature combined. This condition obtains in crowded rooms and auditoriums, because every person gives off both heat and water vapor. Such rooms need primarily new and moderately heated air. Movement of air also helps. Movement under extreme conditions, for example, saturated air of body temperature, would not help heat loss at all; but under usual conditions and for reasons already given, it does help.

Movement of air in our public places and homes should be worked for. If we become accustomed to air in motion, we are less affected by outside conditions, where such movement is the rule. We ought to be accustomed to drafts and not affected by them. I shall not be surprised if we come to use electric fans in winter, even more than in summer.

HUMIDITY IN THE HOME

The greatest problem from the standpoint of humidity is the home or office, where a few people live and work, in winter in our northern states. We take into our houses sub-zero air which has almost no water content and heat it to 70 deg. fahr. Even with the water which such air eagerly laps up from furniture, plants, cooking processes, and from the skin and lungs of every occupant, nevertheless this air in our homes is likely to be dryer than that of any desert on earth.

The body thermostat can adjust the temperature of the internal organs to these conditions. There is not such discomfort and danger as comes from overcrowded unventilated movie theatres or school rooms. But there are somewhat important disadvantages.

In the first place, in order to keep up the body temperature in dry air, much blood is withdrawn from the skin. Moreover evaporation is active. The skin is cooled, and we "feel" chilly. Hence the tendency is to have hotter rooms, up to 75 to 80 deg., and even 85 deg. fahr. The skin gets cracked and rough, which is not pleasant, to say the least. The same tendency to rapid drying extends to the mucous surfaces of the nasal passages, the pharynx and trachea, with consequent respiratory disturbances. Nose and throat specialists, generally, attribute the frequency of infections in winter more to the dryness than to the cold. This all amounts to saying that in order to protect its indispensable internal temperature, the body has to abandon more or less the outlying provinces such as the skin and mucous surfaces. They have not enough blood and get dry; they pick up germs and become inflamed.

More theoretical, as yet, but perhaps to be found of more fundamental importance is the effect on the nervous system, which in the hot stagnant air of our winter homes is assumed to be deprived of the constant tonic of nerve impulses coming from a skin stimulated by cool, moving air.

I advocate strongly the artificial humidification of dwelling houses in winter in our northern states. The standard should be as high as can be secured without precipitation on inside walls. I do not think we can go above 50 per cent in our houses, as ordinarily built, even with double windows.

On account of leakage the amount of water to be evaporated is large; say 15 or 20 gal. a day for a small house. Therefore, the devices on the market, to be used on radiators, are all absolutely inefficient. Tests show that few of them evaporated as much water as one person gives off from his lungs. Most of the devices used in connection with hot-air furnaces are of little real use.

The problem can be solved, but it requires special attention to certain factors of evaporation, namely, surface of contact of water and air and movement of air. These are more important under house conditions than temperature. I have described various experimental humidifiers that are effective. A system to be of much use must be automatic, as few people will carry half or two-thirds of a barrel of water daily to hand-filled humidifiers.

As a matter of opinion I may say that some type of humidifier by which air movement would be created, e. g., a combination of electric fan and convenient water surfaces, would be the ideal. Why should not people be willing to pay for humidity and air movement, two important hygienic factors? I understand, however, that such apparatus has not so far proven a commercial success.

No. 591

A STUDY OF THE INFILTRATION OF AIR IN BUILDINGS

Progress Report upon Tests Carried out Jointly by A. S. H. & V. E. Research Laboratory and the U. S. Bureau of Mines

By O. W. Armspach, Pittsburgh, Pa.
Member

INTRODUCTION

It is the object of this investigation to determine the quantity of air that is infiltrated through the various exterior walls and around windows under the conditions as are generally found in existing buildings. The present is the first in a series of such investigations and this statement in general will consider the apparatus needed to conduct the tests, the methods of conducting the tests, formulæ and calculations and data obtained in the preliminary experiments. A complete study must of necessity be extended to include a large number of existing buildings, so that the coefficients finally derived will apply to those types of constructions commonly found in practice. Natural interchange of air through the interstices in the walls and around windows is due largely to the pressure of the wind on the outside surface. This interchange is maximum when the wind velocity is maximum and perpendicular to the surface, and it is minimum when the wind velocity is minimum and parallel to the surface. Leakage will vary between these two extremes according to the value of these factors. In addition, the height of the building may cause a "chimney" effect on account of the difference in temperature of the air indoors and outdoors. This effect is probably very slight and it is largely destroyed by the action of the wind which has practically the same effect at various points above the ground.

The tests conducted at the present time indicate that the quantity of air entering the room does not vary directly with the wind velocity, but as the wind velocity increases the leakage proportionately decreases. From a theoretical consideration of the problem, this is to be expected since the

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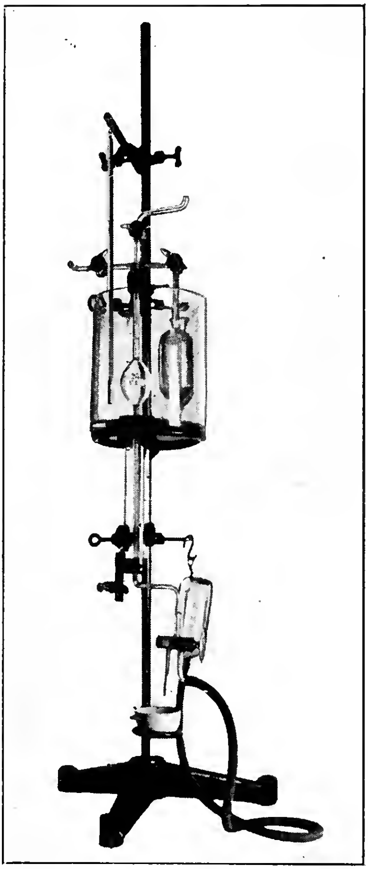

FIG. 1. PETERSON-PALMQUIST GAS ANALYSIS MACHINE PREPARED FOR CALIBRATION

velocity of air through small orifices varies as the square root of the pressure, and hence the volume of air entering the building through the multiplicity of these orifices which we may consider to make up the walls, must vary as the square root of the wind pressure. Furthermore, as the wind pressure increases the window tends to be forced against its "seat," thereby decreasing the size of the crack and in turn the amount of air entering through the crack is decreased.

METHOD OF DETERMINING AIR CHANGE

In determining the infiltration factors for various constructions, the total quantity of air is considered to be composed of two components; first, the quantity of air leakage per square foot of exposed wall surface, and secondly, the quantity of air leakage per linear foot of window crack. Both of these quantities will be determined for various constructions and wind velocities in the tests to follow. It is quite likely that the quantity of air passing directly through the wall is extremely small and if, in view of more complete tests, this is found to be so, the first component can be neglected.

To make the tests, carbon dioxide gas is generated in a room until there will be about 20 parts of CO_2 per 10,000 parts of air. Air samples are then taken every hour for four hours and an analysis for CO_2 is made with the Peterson-Palmquist gas-analysis machine. This instrument is accurately calibrated and all CO_2 analyses were corrected before the final computations were made. As the air in the room containing a high CO_2 content becomes diluted by outdoor air of low CO_2 content, the average CO_2 content in the room will decrease from hour to hour, and the rate of decrease is a function of the rate of dilution by the outdoor air.

Experiments show that the CO_2 liberated, diffuses rapidly throughout the room and its distribution has proved entirely satisfactory. Also, since the rate of diffusion of CO_2 in air is slow compared with the velocity of air entering through the cracks around the windows, no loss of the gas not representing air change is probable at this point. So long as the velocity of air through the crevices is greater than the rate of diffusion of the gas, a flow of CO_2 against the direction of flow of incoming air is impossible.

The wind velocities are determined by taking five hourly anemometer readings outdoors, each reading extending over a period of ten minutes. The readings are converted from feet per minute to miles per hour. They are taken at a point opposite the room to be tested and at a sufficient distance to avoid any counter air currents caused by the wind striking the wall of the building. All anemometer readings were corrected by

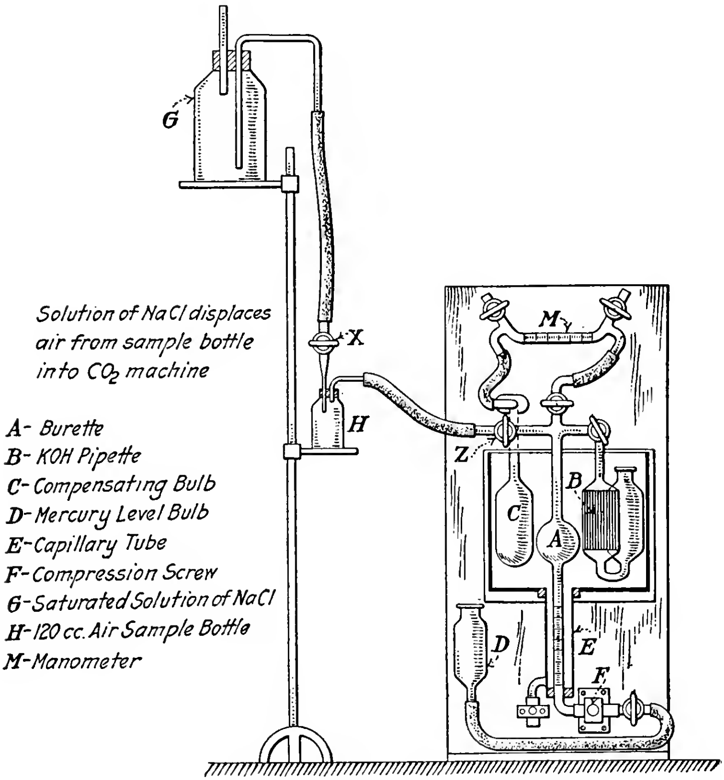

FIG. 2. DIAGRAM OF APPARATUS AND CONNECTIONS OF THE PETERSON-PALMQUIST GAS ANALYSIS MACHINE

referring to a calibration curve of the instrument. The instrument was a Short & Mason with a 2¾ in. diameter of the vanes.

The Peterson-Palmquist machine was found to be best suited to make exact determinations of the CO_2 content of the air. It gives results of CO_2 in parts per 10,000 directly, and although it is very sensitive and requires extreme care in making a determination, it is highly accurate. Each graduation on the machine reads to hundredths of a per cent and thousandths of a per cent are readily estimated. It is important, however, that the instrument should be accurately calibrated. The machine consists essentially of a 25 cc. burette, an absorption chamber containing a KOH solution, a leveling bulb containing mercury, and a manometer and compensating bulb to correct for temperature and pressure changes.

To calibrate the machine it was carefully removed from the frame and fitted to a ringstand as shown in Fig. 1. The rubber tubing connecting the leveling bulb with the burette was removed from the burette. A capillary stop-cock was then fused to the lower end of the burette and the rubber tubing again connected to the lower right arm of the stop-cock. By this arrangement, mercury can be passed into the burette by raising the mercury bulb, and after reversing the stop-cock the mercury can be allowed to pass drop by drop into the porcelain dish which is placed below to receive it. The water jacket was filled and a mercury thermometer was suspended from above to indicate its temperature. The manometer tube not being needed in making the calibration, was removed from the instrument. To determine the total volume of the burette from the zero mark on the scale to the mark above the bulb, the stop-cock was turned so that the mercury bulb communicated with the burette. The mercury was then raised until it stood at the upper mark on the burette, the stop-cock was closed, and the temperature of the water was taken. The stop-cock was then turned, permitting the mercury to slowly run into the previously weighed porcelain crucible. As the level of the mercury approached the lower end of the burette, the stop-cock was partially closed so as to allow the mercury to escape drop by drop, and when its level was exactly on the zero mark, the cock was entirely closed. The weight of the mercury divided by its density at the temperature of the water bath, will give the volume of the burette in cubic centimeters. In like manner, the level of the mercury was brought to the top of the graduated portion marked 80, allowed to flow to the crucible, and weighed at every 10 mark. The following results were obtained:

Weight of crucible empty	= 18.6195 gm.
Weight of mercury plus crucible	= 356.2346 gm.
Total weight of mercury in burette	= 337.6151 gm.
Temperature of the water	= 26 deg. cent.
Density of mercury at 26 deg. cent.	= 13.5315 gm.

$$\text{Therefore the total volume of the burette} = \frac{337.6151}{13.5315} = 24.9503 \text{ cc.}$$

Likewise, considering the volume of the graduated portion of the burette, and its ratio to the total volume, the following corrections were calculated:

Scale Reading	Corrected Reading
10	9.77
20	19.44
30	29.50
40	39.49
50	49.24
60	59.34
70	68.91
80	78.91

After the calibration was completed, the capillary stop-cock which had previously been fused to the lower stem of the burette, was removed, the leveling bulb was connected to the stem, and the whole instrument was mounted on its proper frame to be used for making the analyses. Before being used, however, it is essential that a drop of water be placed in the burette over the mercury column and also a few drops placed in the compensating bulb in order to insure equal vapor pressure in both chambers.

METHOD OF CONDUCTING TESTS

The CO_2 content inside the room tested was brought to the desired point by adding a calculated quantity of sulphuric acid to a determined quantity of sodium bicarbonate. A union of the two compounds will produce CO_2 gas. The chemicals were divided into four equal portions so as to liberate CO_2 at four points in the room; thereby giving a better and more rapid distribution throughout. Sufficient time was allowed after the CO_2 had been generated, to insure proper diffusion throughout the room, this time being usually 30 minutes. The quantity of gas to be generated will depend on the size of the room and the concentration desired. The weight of chemicals needed for various sized rooms is calculated as follows:

$$2NaHCO_3 + H_2SO_4 = Na_2SO_4 + 2H_2O + 2CO_2$$

24.45 litres of CO_2 weigh 44 grams

$$\text{Therefore } 44 \, \frac{28.32}{24.45} = 51.0 \text{ grams } CO_2 \text{ per cu. ft.}$$

The molecular weight of $NaHCO_3 = 84$
The molecular weight of $H_2SO_4 = 98$

$$\text{Therefore } \frac{51.0 \times 84}{44} = 97.4 \text{ grams } NaHCO_3 \text{ per cu. ft. of } CO_2$$

$$\text{Also } \frac{51.0 \times 98}{2 \times 44} = 113.5 \text{ gr. of 100 per cent } H_2SO_4$$

or 118.3 grams of 96 per cent H_2SO_4 per cu. ft. of CO_2

Therefore the chemicals needed to generate 1 cu. ft. of CO_2 are 97.4 grams sodium bicarbonate and 65 cc. sulphuric acid.

The room tested contained approximately 4250 cu. ft., and on a basis of 20 parts of CO_2 per 10,000 the chemicals needed are as follows:

$$\frac{4250 \times 20}{10{,}000} = 8\tfrac{1}{2} \text{ cu. ft. } CO_2 \text{ required}$$

$$8\tfrac{1}{2} \times 97.4 = 827 \text{ grams of } NaHCO_3$$

$$8\tfrac{1}{2} \times 65 = 552 \text{ cc. } H_2SO_4$$

The general formula for any sized room may be given by the equation:

$$CO_2 = \frac{C \times R}{10{,}000} \qquad (1)$$

in which C = cubic content of room
R = parts of CO_2 per 10,000
CO_2 = cu. ft. of gas that must be added to bring the room to the required content.

Four 1 gal. jars were placed on the floor near the corners of the room. In each jar 200 cc. of distilled water was placed, and to this was added one-quarter of the 827 grams of $NaHCO_3$. The doors and transoms were closed and kept so throughout the test. At a stated period, one-fourth of the 552 cc. of the H_2SO_4 was gently poured into each jar and a violent reaction took place, resulting in the liberation of CO_2. The air entering the room through the cracks around the windows found its way out through the cracks around the transom and doors leading to the corridor. After thirty minutes had elapsed, four air samples were taken at points

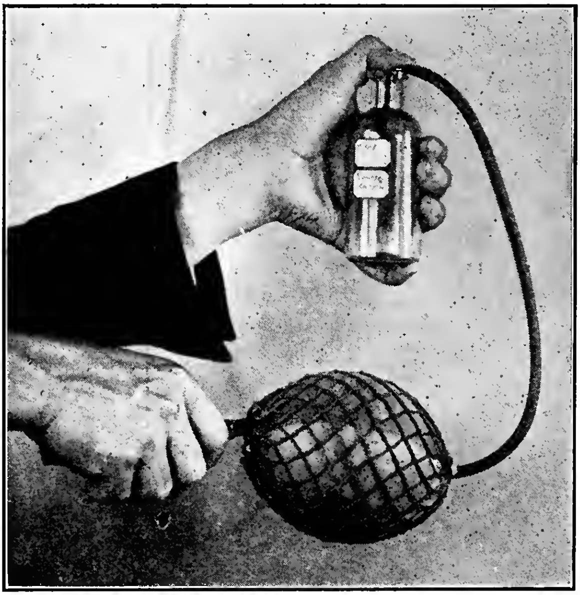

FIG. 3. VIEW SHOWING METHOD OF OBTAINING AN AIR SAMPLE CALIBRATION OF CO_2 MACHINE

4½ ft. from the floor near the corners of the room and also four directly above these, 9 ft. from the floor. Eight samples were obtained in this manner at the beginning and end of each hour for a period of four hours, so that in all, forty samples were obtained. Two analyses were made of each sample, making a total of 80 analyses for each test.

Fig. 3 shows the method followed in taking the samples. The rubber tube attached to the bulbs is inserted to the bottom of the bottle and held at arm's length so that the sample will not become contaminated by expired air. This tube is closed by pressing it between the thumb and neck of the bottle of 120 cc. capacity, and the net covered bulb is filled with air by pressing the uncovered bulb several times with the hand. The thumb is released and the inrushing air replaces the air which was originally in the bottle. This operation is repeated three times to insure that the bottle is entirely filled with the air in the room. The tube is removed and the bottle is tightly sealed with a rubber stopper and taken to the laboratory for analysis. Each bottle is carefully marked so as to indicate the exact time when the sample was taken and also its location. In addition to the samples taken in the room, two were obtained outdoors at the start of the test and also two at the end. A record of the temperature and humidity outdoors and indoors was also obtained.

To make the analysis, the rubber stoppers are removed under a saturated solution of NaCl, and the opening is covered with the finger. The bottle (see H in Fig. 2) is then attached to the stopper holding a tube leading to G which contains a saturated solution of NaCl. By opening stop-cocks x and z and lowering the mercury bulb, the NaCl solution

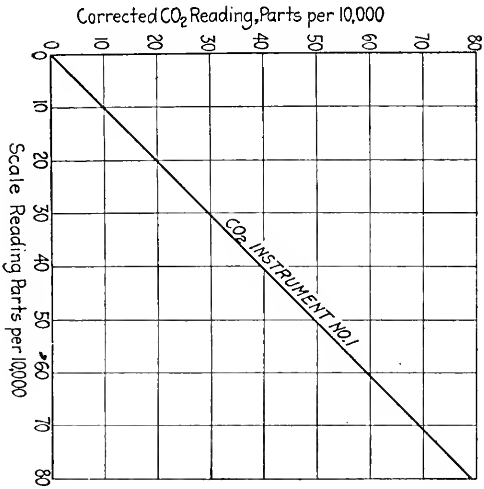

FIG. 4. CALIBRATION CURVE OF PETERSON-PALMQUIST GAS ANALYSIS MACHINE

flows into the sample bottle and drives the air into the burette of the machine where the analysis is made for CO_2.

FORMULA FOR DETERMINING AIR QUANTITIES

The sixteen analyses made for the CO_2 content at the beginning of the hour were corrected in accordance with the calibration curve and then an average for the room was obtained. This corrected average gives the drop in the CO_2 content during an hour due to dilution with outdoor air, and from these values, the quantity of air infiltrated may be determined.

Let $S =$ total cu. ft. CO_2 from any source in room;

$S = 0.6 \times n$ ($n =$ number of occupants);

$(CO_2)_1 = CO_2$ in outside air, parts per 10,000;

$(CO_2)_4 = CO_2$ in room at beginning of hr., parts per 10,000;

$(CO_2)_5 = CO_2$ in room at end of hr., parts per 10,000;

$Q_1 =$ cu. ft. of air entering per hr.;

$Q_2 =$ cu. ft. of air leaving per hr.;

$C =$ cu. ft. of room capacity;

Then:

$$\frac{(CO_2)_1 Q_1}{10{,}000} = \text{cu. ft. of } CO_2 \text{ entering per hr.;}$$

$$\frac{(CO_2)_4 C}{10{,}000} = \text{cu. ft. of } CO_2 \text{ in room at start of hr.;}$$

$$\frac{(CO_2)_5 C}{10{,}000} = \text{cu. ft. of } CO_2 \text{ in room at end of hr.;}$$

$$\left(\frac{(CO_2)_4 + (CO_2)_5}{2}\right) \frac{Q_2}{10{,}000} = \text{cu. ft. } CO_2 \text{ leaving the room per hr.;}$$

Also $Q_1 = Q_2$

Therefore
$$\frac{C(CO_2)_4}{10{,}000} + \frac{Q_1(CO_2)_1}{10{,}000} + S = \frac{C(CO_2)_5}{10{,}000} + \frac{Q_2}{10{,}000}\left[\frac{(CO_2)_5 + (CO_2)_4}{2}\right]$$

And
$$C(CO_2)_4 + Q_1(CO_2)_1 + 10{,}000 \times S = C(CO_2)_5 + Q_1\left[\frac{(CO_2)_5 + (CO_2)_4}{2}\right]$$

$$C\ [(CO_2)_4 - (CO_2)_5] + 10{,}000 \times S = Q_1\left[\frac{(CO_2)_5 + (CO_2)_4 - 2(CO_2)_1}{2}\right]$$

Hence:
$$Q_1 = \frac{2C\ [(CO_2)_4 - (CO_2)_5] + 20{,}000 \times S}{(CO_2)_5 + (CO_2)_4 - 2(CO_2)_1} \qquad (2)$$

In order to separate the portion of air passing through the walls from the portion passing through the cracks around the windows, tests will be made in two rooms in the same building and having a different ratio of window to exterior wall surface. The calculations for the quantity of air entering at each point may then be made in accordance with the equation: $Q = wx + gy$

(3)

in which Q = total cu. ft. of air entering the room;
w = outside wall surface in sq. ft.;
x = cu. ft. of air passing through the wall per sq. ft.;
g = linear ft. of window crack;
y = cu. ft. of air entering through each foot of crack.

By making tests in two rooms and substituting the values of Q obtained in each test into equation (3), x and y may be calculated.

PRELIMINARY RESULTS AND DATA

The preliminary tests were conducted in the Bureau of Mines building in a room on the second floor having a cubic content of 4178 cu. ft. including a deduction for space occupied by furniture. The room is approximately 14 ft. 6 in. by 21 ft. by 13 ft. 8 in. The exterior wall is

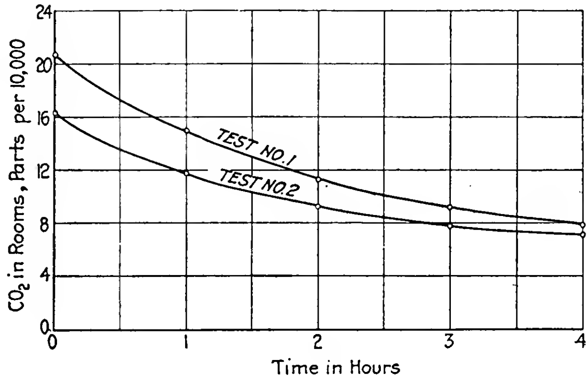

FIG. 5. CURVE SHOWING DECREASE IN CO_2 IN ROOM DUE TO AIR INFILTRATION

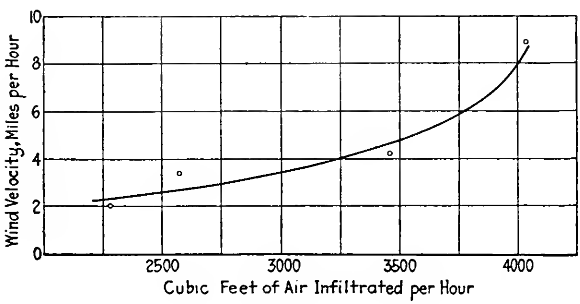

FIG. 6. CURVE SHOWING EFFECT OF WIND VELOCITY ON INFILTRATION IN FOUR TESTS OF A ROOM

built of 18 in. of brick plastered on the inside. The windows are the sliding-sash type and all sliding portions are fitted with metal weather strips. The construction would be considered very tight and of the best workmanship. The exterior wall includes 102 sq. ft. of wall surface, 96 sq. ft. window surface and 46 ft. 1 inch of window crack. The room has three doors, one on either side leading to the adjacent offices and one in the wall opposite the window leading to the corridor. Above the latter is a 42 in. by 18 in. hinged transom. There are sufficient cracks around the doors and transom to permit air to leave the room as it is forced out

TABLE 1. RESULTS OF CO_2 ANALYSES, TEST NO. 2

Room 277—Nov. 26, 1920

Sample No.	Time	1st Sample 1st Read.	1st Sample Check	2nd Sample 2nd Read.	2nd Sample Check	Average	Corrected Average
1	9:45	17.9	17.8	17.3	17.2	17.550	
2		19.1	18.9	18.8	19.1	18.975	
3		18.2	18.1	17.8	18.1	18.050	
4		13.4	15.3	15.0	15.0	15.175	
5		16.0	16.1	15.9	16.0	16.000	
7		16.4	16.2	16.3	16.3	16.300	
8		16.0	16.5	16.0	16.1	16.150	
					Average	16.9	16.342
9	10:45	13.1	13.5	12.4	12.6	12.900	
10		11.9	12.0	11.5	11.2	11.650	
11		12.1	12.8	11.8	11.6	12.075	
12		12.9	12.6	13.0	12.3	12.750	
13		12.2	12.0	12.0	12.1	12.075	
14		11.0	11.0	11.0	11.0	11.000	
15		13.0	13.0	12.3	12.6	12.725	
16		12.0	11.5	11.2	11.3	11.500	
					Average	12.08	11.781
17	11:45	10.2	10.7	9.8	10.4	10.275	
18		10.3	10.2	10.0	9.8	10.075	
19		10.2	10.3	9.0	9.0	9.625	
20		9.4	9.4	9.0	9.1	9.225	
21		9.1	9.2	9.7	9.2	9.300	
22		9.3	9.3	9.8	9.7	9.525	
23		9.6	9.5	9.1	9.2	9.350	
24		8.6	8.2	8.9	8.9	8.650	
					Average	9.45	9.233
25	12:45	8.0	8.1	7.7	7.8	8.400	
26		8.0	8.1	8.1	8.0	8.050	
27		8.2	8.2	8.4	8.2	8.250	
28		8.4	8.2	8.0	8.0	8.150	
29		8.2	8.1	7.7	7.7	7.925	
30		7.3	7.4	7.1	6.8	7.150	
31		7.6	7.1	7.5	7.2	7.350	
32		7.9	7.7	7.9	7.9	7.850	
					Average	7.89	7.703
33	1:45	8.2	8.5	8.1	7.7	8.125	
34		7.8	7.2	7.7	7.1	7.450	
35		7.5	7.3	7.2	7.2	7.300	
36		7.7	7.3	7.2	7.2	7.325	
37		7.4	7.3	7.3	7.3	7.325	
38		7.8	7.3	7.3	7.8	7.550	
39		6.0	5.9	6.2	6.2	6.075	
40		7.2	7.0	7.3	7.2	7.175	
					Average	7.29	7.112
Outdoor Samples	9:15	4.1	4.1	4.3	4.0	4.225	
		4.2		3.8		4.000	
	1:45	4.2	4.0	4.2	4.4	4.200	
		4.1	3.9	4.0	3.9	3.975	
					Average	4.100	4.00

Sample No. 6 was contaminated.

TABLE 2. SUMMARY OF FOUR TESTS

Test number	1	2	3	4
Wind velocity, miles per hr.	2.0	3.4	4.2	8.9
*Wind direction	30°	30°	30°	30°
Infiltration per hr. in cu. ft.	2285	2577	3461	4032
Cu. ft. air per foot of crack per hr.	49.5	55.8	75.2	87.6
Air change per hr.	0.54	0.61	0.83	0.97

* Refers to angle the direction of the wind makes with the wall surface.

by air entering through the exterior wall. Table 1 gives the results of the analysis of test No. 2. By substituting the corrected average for the end of each hour into equation (2), the following results were obtained:

TABLE 3. AVERAGE RESULTS OF TESTS

Time	Wind Vel., miles per hr.	CO_2 drop	Cu. ft. air entering
9:45 to 10:45	3.15	4.561	2491
10:45 to 11:45	3.33	2.548	2560
11:45 to 12:45	3.55	1.530	2766
12:45 to 1:45	3.65	0.591	2491
Average	3.37		2577

Wind direction	30 deg. to wall
Average wind velocity	3.37 miles per hr.
Average air infiltration	2577 cu. ft. per hr.
Cu. ft. air per foot of crack	55.8 cu. ft. per hr.

In the reports to follow, the results of the tests conducted in the various classes of buildings and in buildings embodying the different types of construction under varying wind velocities will be given. With this information, infiltration factors may be prepared that will permit by computation, the determination of the exact amount of radiation that must be allowed for inleakage of air through the different constructions and also the determination of the quantities of outdoor air available in naturally ventilated rooms.

ACKNOWLEDGMENT

The author acknowledges his indebtedness to Louis Ebin of the Research Bureau who made the analyses of the large number of air samples necessary in these tests.

DISCUSSION

PERRY WEST: Have these results been compared in any way with the direct measurement results, from tests made some time ago, that is, by the boxed-in window plan, of measuring the air coming in?

THE AUTHOR: We tried to do this and I think in only one case did we find that we could at all compare the results. The difficulty is that those tests were made at wind velocities that we do not meet with outside. It seems that their curves are for winds away above the velocities ordinarily met with. We did try to extend the curves obtained in laboratory experiments and found that the results closely approximated those obtained in our experiments, but an accurate comparison cannot be made. This is the reason we conducted tests in existing buildings, so that we could get the influence of the actual outdoor conditions.

No. 592

TRANSMISSION OF HEAT THROUGH SINGLE-FRAME DOUBLE WINDOWS

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Non-Member

THE experiments described in this article were performed as the result of an inquiry from F. W. Dakin of Sherbrooke about the relative transmissions of heat for single-frame double windows of various thicknesses. It was thought that an outline of the method of attack and an account of the experiments should be recorded, because the existing data available for handling this particular problem appear to be conflicting and unreliable, and the matter is one of considerable interest to engineers and architects.

Two sizes of single-frame double windows were examined, and they are referred to subsequently as the "half inch" and "three-quarter inch" windows respectively. The actual dimensions are tabulated below, and it will be noted that they differ from these specified values.

In order to eliminate the common errors due to external influences in such experiments, the apparatus shown in Fig. 1 was constructed. In each case there are six parallel plates of glass at the recorded distances apart, separated by wooden frames and thin rubber strips. These were clamped together and gave a convenient water-tight apparatus which could be pulled to pieces and reassembled with ease. Each of the five compartments could be continually supplied with water or air by means of tubes let into the frames; and other holes were provided for thermometers and stirrers.

Further descriptive details are given in the record of experiments. The compartments are referred to as Nos. 1, 2, 3, 4, and 5, being numbered consecutively from one side to the other. In the tests, the transmission of heat through the air-filled compartments, No. 2 and No. 4, was determined by observations on the cooling or warming of the water-filled centre compartment, No. 3, while the outer water-filled compartments, Nos. 1 and 5, were kept at a constant temperature.

Owing to the constant breakage of the windows on account of unequal heating and clamping, less than half of the experiments commenced could be carried to a successful conclusion.

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THE DIMENSIONS OF THE COMPARTMENTS

A series of repeated measurements gave the mean results for the dimensions of the compartments (see Table 1). The figures (such as 3—1, etc.) indicate how some of the values are determined from the others.

TABLE 1. MEASUREMENT FOR A COMPARTMENT IN A VERTICAL POSITION

	½ in. Frames cm.	in.	¾ in. Frames cm.	in.
1. Average total thickness when full of air.....	2.004	0.789	2.435	0.958
2. Maximum total thickness when full of water.	2.538	0.999	2.983	1.174
3. Average total thickness when full of water..	2.271	0.894	2.709	1.066
4. Average thickness of "bulge" produced by the weight of water (3-1)........................	0.267	0.105	0.274	0.108
5. Thickness of the two panes of glass.........	0.426	0.168	0.396	0.156
6. Average internal thickness (l) when full of water (3-5)	1.845	0.726	2.313	0.411
7. Average internal thickness of adjacent air compartment (1-4-5)	1.311	0.516	1.765	0.694
8. Average internal thickness, when each is full of air (1-5)	1.578	0.621	2.039	0.802
9. Inside height of frame......................	40.80	16.07	40.85	16.09
10. Inside width of frame......................	35.70	14.06	35.75	14.08

TABLE 2. VOLUMES OF COMPARTMENTS, WHEN FILLED WITH WATER

		cu. cm.
½ in. Compartment	By calculation $40.80 \times 35.70 \times 1.845 =$	2688
	Mean of three direct measurements with a graduated vessel	2680
¾ in. Compartment	By calculation $40.85 \times 35.75 \times 2.313 =$	3378
	Mean of three direct measurements with a graduated vessel	3390

For the purpose of the subsequent calculations, No. 6 should be the most accurately determined. As a check the volume of water required to fill a compartment was measured independently with the close agreement shown in Table 2.

The bulging of the apparatus when filled with water thus leads to an uncertainty of not more than a third of one per cent in the determination of the thicknesses (l) of the compartments. (1.845 cm. and 2.313 cm. respectively in our apparatus.)

PRELIMINARY CALCULATION OF THE APPROXIMATE VALUE OF THE TRANSMISSION OF HEAT

If the customary methods of calculation employed in heating and ventilating problems are applied to the single-frame double window, an approximate but somewhat doubtful value is obtained. It becomes apparent, however, that the difference between a ½ in. and a ¾ in frame should be negligible.

Let us consider a single-frame double window through which there is a steady flow of heat:

Where t_1 = temperature of the warm air coming in contact with the first pane;

t_2 = temperature of the glass, "just inside" the outward surface of the first pane;

t_3 = temperature of the glass "just inside" the inward surface of the first pane;

t_4 = temperature of the air "just inside" beyond the first pane;

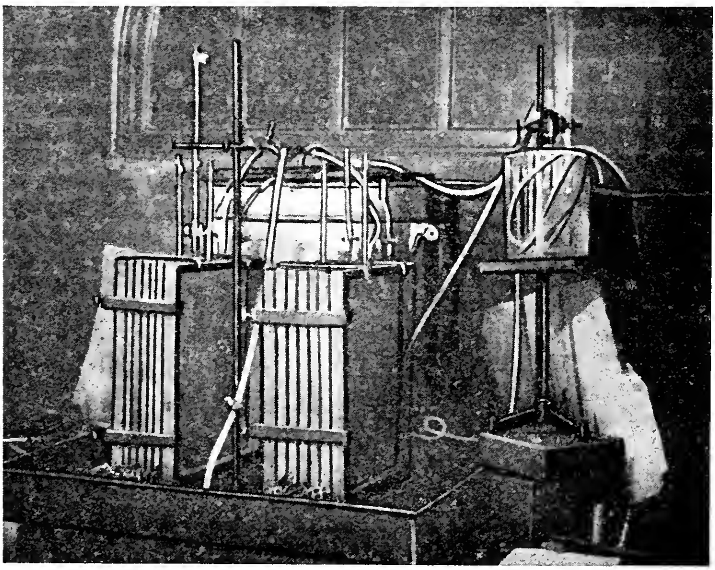

FIG. 1. VIEW OF THE TEST APPARATUS FOR HEAT TRANSMISSION THROUGH SINGLE-FRAME DOUBLE WINDOWS

t_5 = temperature of the air "just inside" before the second pane;

t_6 = temperature of the glass "just inside" the inward surface of the second pane;

t_7 = temperature of the glass "just inside" the outward surface of the second pane;

t_8 = temperature of the cold outer air supposed to be coming rapidly into contact with the second pane;

d = thickness of each pane of glass;

x = distance between the panes;

c = specific thermal conductivity of the glass;

a_1, a_2, a_3 and a_4 = equivalent total conductivities per sq. unit of the thin boundary regions respectively at the first, second, third and fourth planes of contact between the air and the glass;

k = equivalent conductivity per sq. unit of the whole window;

X = equivalent conductivity per sq. unit, per unit length of thickness, of the confined air space which is bounded on each side by the *thin regions* of air which are in contact with the glass and which are involved in the quantities a_2 and a_3. (X includes part of the convection effects, is a comparatively large quantity, may vary with the dimensions, and must not be confused with the low specific conductivity of air.)

Most of these quantities cannot easily be ascertained; and in view of the fact that the largest temperature gradients are in the thin regions of air which are in the immediate vicinity of the four surfaces of contact between the air and the glass, it is apparent that the interpretation and the definition of the phrases *just inside* and *thin region* are very difficult to achieve. For an approximate solution, however, it will be sufficient for us to assume that these quantities may have equivalent empirical values.

In a steady state of flow, it is obvious that

$$k(t_1-t_8)=a_1(t_1-t_2)=\frac{c}{d}(t_2-t_3)=a_2(t_3-t_4)=$$

$$\frac{X}{x}(t_4-t_5)=a_3(t_5-t_6)=\frac{c}{d}(t_6-t_7)=a_4(t_7-t_8).$$

Eliminating the temperatures, this can be reduced to

$$\frac{1}{k}=\frac{1}{a_1}+\frac{d}{c}+\frac{1}{a_2}+\frac{x}{X}+\frac{1}{a_3}+\frac{d}{c}+\frac{1}{a_4}$$

According to the data in engineering texts, the values for a for air-glass contacts, range from about 1.5 to 2.1 B.t.u. per sq. ft. per hr., per deg. fahr. difference. (See for example, *Heating and Ventilation,* by Allen and Walker, p. 14.) The lower value corresponds to the conditions when the air is stagnant and the total temperature difference small, while the higher value corresponds to the conditions when the motion of the air is rapid and the total temperature difference large. In the case of thin single-frame double windows, the convection inside is brisk, and although the motion of the air will be usually less than that outside, we may take 2.1 as the upper limit for transmission. (This value is probably still too low for a for the outer surface on days of strong cold winds, and indeed further experimental work will probably show that it is too low in any case. This is probable because the effect of x/X has usually been ignored and thus really included in the a's, which means that the empirical values of a have been recorded as less than they should be.)

For glass c is about 0.55, hence for panes about 0.08 in. thick, $2d/c=$ 0.024 B.t.u. per sq. ft. per hr., per deg. fahr. difference. The formula for maximum transmission under practical conditions then reduces to

$$1/k=1.91+0.024+x/X$$

If we neglect x/X this gives for the whole window, $k=0.52$ B.t.u. per sq. ft., per hr., per deg. fahr. difference. (If x/X is not negligible, it will reduce the magnitude of k in the last equation, but if the a's were originally determined in such a way that an x/X effect was included under different circumstances, the value of k corrected, could be greater.)

The author estimates that X should probably be given a variable value much greater than 3.0 per in. If we take $X=3.0$, we get for the ½ in. double window, $k=0.48$ and for the ¾ in. double window, $k=0.46$ B.t.u. per sq. ft., per hr. per deg. fahr. difference.

The figures in the last two paragraphs have been derived chiefly to indicate that the transmission of heat through these windows would be expected to vary very gradually with the thickness. The calculations are not submitted as an accurate estimate of the absolute value of the transmission. The experiments described below, indicate that the values of the empirical a's are probably on the average nearer to 3 than to 2 for the air-glass contacts in the case of the particular windows in question, and that the correct values of k for these windows lie between 0.60 and 0.70 B.t.u. per sq. ft. per hr., per deg. fahr. difference.

THE DETERMINATION OF THE CONDUCTIVITY BY COOLING AND WARMING EXPERIMENTS

Preliminary experiments led to curves for *cooling* of an exponential character, and confirmed the expectation that Newton's law of cooling would hold approximately for the central compartment in these tests. It is apparent therefore, that due weight can be given most readily to all the observations by dealing with the linear graphs which can be obtained and analyzed as follows:

Let $\theta_c =$ average temperature (deg. cent.) of the outer compartments, which is maintained approximately constant;

$\theta =$ temperature (deg. cent.) of the central compartment at a time t (min.) after the observations have commenced;

$\delta q =$ average quantity of heat (calories) which flows to or from the central compartment, per sq. cm. in a short interval δt min.;

$m =$ mass (gm.) of water in the central compartment;

$a =$ area (sq. cm.) of each window;

$l =$ internal thickness (cm.) of the central compartment;

$k' =$ average amount of heat (calories) which would flow across one of the air compartments per sq. cm., per min., per deg. cent. difference between the cold and hot compartments (this is approximately a constant over the range of temperatures considered. It is sixty times the conductivity as usually expressed per sec.).

Applying Newton's cooling law, neglecting the effects at the frames, and taking the density of water as unity, we have:

$$\frac{dq}{dt} = k' \, (\theta - \theta_c) = \frac{m}{2a}\frac{d\theta}{dt} = \frac{l}{2}\frac{d\theta}{dt}$$

The factor, 2, arises to account for the losses from each side; solving the above, we get: $\theta - \theta_c = Ae^{\underline{|2k't/l}}$

where A is the difference in temperature when $t = 0$.

If t is plotted as abscissa and the $\log_{10} (\theta - \theta_c)$ as ordinate, then the equation of the linear graph which will be obtained for a series of observations, is: $2.303 \log_{10} (\theta - \theta_c) = 2k't/l + 2.303 B$; where $B = \log_{10} A$ is the intercept on the ordinate axis of logs.

The value of k' which is the characteristic sought in these experiments, can be calculated at once as follows, by noting also the intercept on the axis of t. If T is the value of this intercept in a particular case, and if L is the value of $\log_{10} (\theta - \theta_c)$ where the axis of t has been drawn, then $2.303 L = 2k'T/l + 2.303 B$

Therefore: $k' = 1.151 \, l(L-B)/T$ calories per min.

In engineering tables which relate to allied problems, k' is usually expressed in B.t.u. per sq. ft. per hr. per deg. fahr. difference. If k is the value in these units and l, T, L and B are obtained as above, we have $k = 123.0 \, k' = 141.6 \, l(L-B)/T$ B.t.u. per hrs.

By obtaining L, B, and T from a linear graph in this manner, the repetition of calculations over the range of observations in order to eliminate temperature errors, is avoided.

RECORD OF EXPERIMENTS AND RESULTS

Experiment No. 1. The central compartment (No. 3) of the apparatus with ¾ in. frames, was filled with hot water and allowed to cool under observation, while cold water was flowing through the outer compartments (Nos. 1 and 5) at such a rate that their temperatures remained approximately constant. The compartments were placed in the vertical position in all of these tests.

In this experiment, the hot compartment was not stirred, but the necessary correction was obtained experimentally, and checked again in comparison with some of the later tests in which there was regular stirring before each reading of the temperature.

In Table 3, the column headed t represents the time in minutes from the commencement of the observations; θ_1 and θ_5 represent the temperatures, cent., in the cold compartments, and θ_3 the temperature in the hot compartment. The column headed $\theta_3 - \theta_c$ represents the difference between θ_3 and the average temperature, approximately constant, of the cold compartments; it gives therefore the "drive." The last column gives the log of $\theta_3 - \theta_c$ to the base 10.

TABLE 3. FOR THREE-QUARTER INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_3-\theta_c)$ deg. cent.	$\log_{10}$ $(\theta_3-\theta_c)$
0	18.7	17.0	40.9	23.4	1.369
5	18.7	17.0	40.2	22.7	1.356
15	18.7	16.8	38.65	21.15	1.325
27	19.0	16.5	36.9	19.4	1.288
37	19.0	16.5	35.7	18.2	1.260
47	19.0	16.5	34.4	16.9	1.228
57	19.0	16.5	33.3	15.8	1.199
72	18.0	16.0	31.9	14.4	1.158
92	18.0	16.0	29.9	12.4	1.093
122	17.0	16.0	27.5	10.0	1.000
132	17.0	16.0	26.8	9.3	0.968

Average value of θ_c is 17.5 deg.

In Fig. 2 the cooling curve is shown; the dotted lines are drawn merely to indicate at once the curvature of the former. In Fig. 3 it can be seen that the logarithmic plotting gives a good linear result, demonstrating the exponential character of the cooling and thus the exactitude with which Newton's law of cooling has been obeyed.

In order to correct approximately for the lack of stirring, observations on the vertical temperature "gradient" in the compartments were made. It was found that the top and bottom of the cold compartments remained at equal temperatures throughout on account of the flow of the water, but in the hot compartment it was found that the top was 3.2 deg. hotter than the bottom after it had been cooling for 92 min. As the whole compartment was at an equal temperature at the start, it was assumed that the final reading, if the compartment had been stirred, would have been approximately 1.6 deg. lower. The dotted line represents the corrected graph.

It can be seen from the graph that $B = 1.37$, $T = 127$ and $L = 0.900$. As already explained the equivalent value of l for the "three-quarter inch" apparatus was taken as 2.313 cm.; hence:

$k = 141.6 \times 2.313 \times 0.47/127 = 1.21$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

Experiment No. 2. In this experiment, the apparatus with the "half inch frames" was used, and a set of observations was obtained in the same manner as in experiment No. 1. As before, a correction for non-stirring was necessary.

Fig. 4 gives the graph. In this case the correction for non-stirring was approximately 2 deg. at 90 min., and leads to the change in slope which is indicated. It can be seen that $B = 1.412$, $T = 105$, $L = 0.900$, and as already explained, l for the ½ in. frames was taken as 1.845 cm.; hence: $k = 141.6 \times 1.845 \times .512/105 = 1.27$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

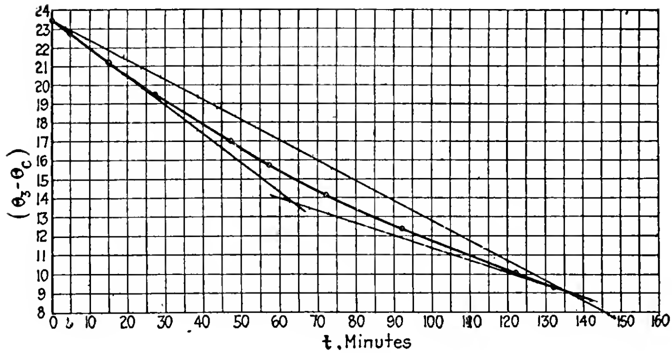

Fig. 2.

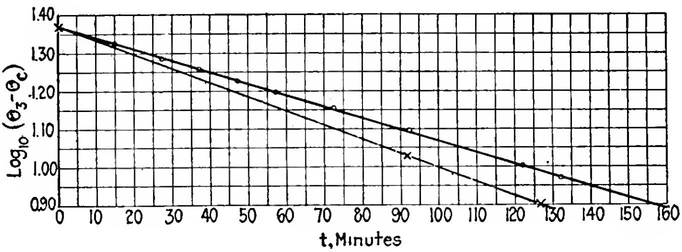

Fig. 3.

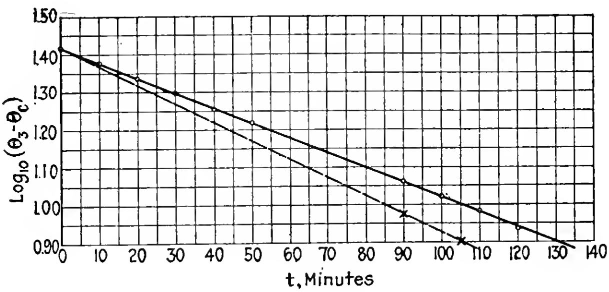

Fig. 4.

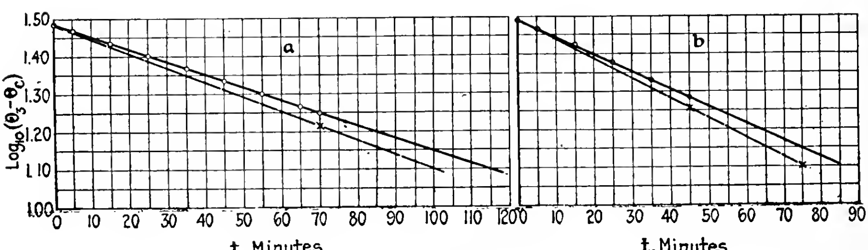

Fig. 5

FIGS. 2-5. CHARTS SHOWING RESULTS OF THE VARIOUS EXPERIMENTS

TABLE 4. FOR HALF-INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_3-\theta_c)$ deg. cent.	$\log_{10}$ $(\theta_3-\theta_c)$
0	16.0	16.0	41.5	25.8	1.412
10	16.0	16.0	39.4	23.7	1.375
20	16.0	16.0	37.35	21.65	1.336
30	16.0	16.0	35.5	19.8	1.297
40	16.0	16.0	33.75	18.05	1.257
50	16.0	16.0	32.3	16.6	1.220
90	15.0	15.0	27.3	11.6	1.065
100	15.0	15.0	26.25	10.55	1.023
110	15.0	15.0	25.3	9.6	0.982
120	15.0	15.0	24.3	8.6	0.935

Average value of θ_c is 15.7 deg.

Experiment No. 3. In this test, experiments similar to Nos. 1 and 2 were performed simultaneously. The results are given in Table 5.

The corrections for non-stirring were approximately 0.9 deg. cent. at 70 min. for (*a*), and 1 deg. cent. at 45 min. for (*b*) as shown in Fig. 5

Thus for (*a*), $B = 1.384$, $T = 100$, $L = 1.000$ and $l = 2.313$; hence $k = 141.6 \times 2.313 \times 0.384/100 = 1.26$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

And for (*b*), $B = 1.387$, $T = 75$, $L = 1.000$ and $l = 1.845$; hence $k = 141.6 \times 1.845 \times 0.387/75 = 1.35$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

Experiment No. 4. The two pieces of apparatus were operated again simultaneously, but on this occasion stirring rods had been placed in the

TABLE 5. (*a*) FOR THREE-QUARTER INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_3-\theta_c)$ deg. cent.	$\log_{10}$ $(\theta_3-\theta_c)$
0	15.0	15.25	39.3	24.2	1.384
5	15.0	15.2	38.4	23.3	1.367
15	15.0	15.2	36.8	21.7	1.337
25	15.0	15.2	35.0	19.9	1.299
35	15.0	15.2	33.45	18.35	1.264
45	15.0	15.2	32.0	16.9	1.228
55	15.0	15.1	30.7	15.6	1.193
65	15.0	15.3	29.7	14.6	1.164
70	15.0	15.2	29.0	13.9	1.143

Average θ_c = 15.1 deg.

TABLE 5. (*b*) FOR HALF-INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_3-\theta_c)$ deg. cent.	$\log_{10}$ $(\theta_3-\theta_c)$
0	15.5	15.05	39.6	24.55	1.387
5	15.5	15.0	38.2	22.95	1.361
15	15.5	15.0	36.3	21.05	1.323
25	15.5	15.0	34.07	18.82	1.275
35	15.5	15.0	32.11	16.86	1.227
45	15.5	15.0	30.5	15.25	1.183

Average θ_c = 15.25 deg.

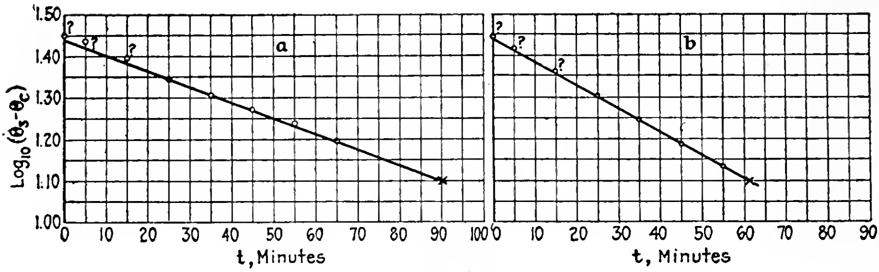

Fig.6.

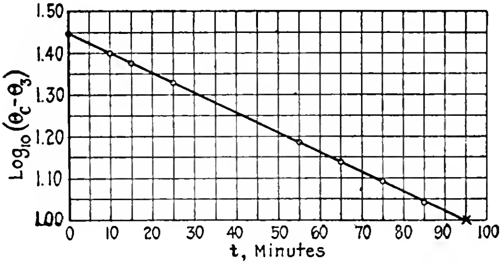

Fig.7

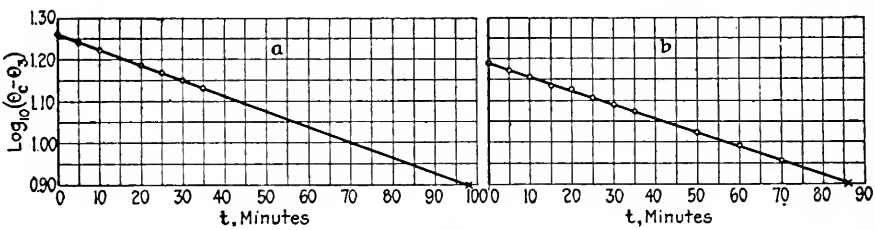

Fig.8.

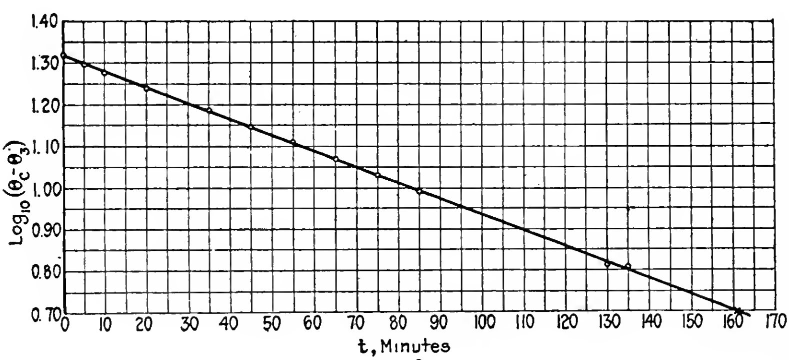

Fig.9

FIGS. 6-9. CHARTS SHOWING RESULTS OF THE VARIOUS EXPERIMENTS centre compartments and were used for half a minute before each observation. The results are shown in Table 6 and in Fig. 6, no corrections for non-stirring being necessary in these cases.

For (*a*), $B = 1.432$, $T = 90$, $L = 1.10$ and $l = 2.313$; hence: $k = 141.6 \times 2.313 \times 0.332/90 = 1.21$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

And for (*b*), $B = 1.438$, $T = 61$, $L = 1.10$ and $l = 1.845$; hence: $k = 141.6 \times 1.845 \times 0.338/61 = 1.45$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

TABLE 6. (*a*) FOR THREE-QUARTER INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_3-\theta_c)$ deg. cent.	$\log_{10}$ $(\theta_3-\theta_c)$
0	..	..	43.6	28.3	1.452
5	..	..	42.6	27.3	1.436
15	*	*	39.8	24.5	1.389
25	15.2	15.6	37.5	22.2	1.346
35	15.2	15.5	35.6	20.3	1.308
45	15.0	15.5	34.0	18.7	1.272
55	15.0	15.4	32.4	17.1	1.233
65	15.0	15.4	31.0	15.7	1.196

Average for $\theta_c = 15.3$ deg.

TABLE 6. (*b*) FOR HALF-INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_3-\theta_c)$ deg. cent.	$\log_{10}$ $(\theta_3-\theta_c)$
0	..	..	43.1	27.9	1.446
5	..	..	41.7	26.5	1.423
15	..	..	38.2	23.0	1.362
25	*	*	35.2	20.0	1.301
35	15.5	15.2	32.8	17.6	1.246
45	15.5	15.0	30.6	15.4	1.188
55	15.4	15.0	28.8	13.6	1.136
65	15.3	14.9	27.1	11.9	1.076

Average for $\theta_c = 15.2$ deg.

* The first three or four readings in these cases were uncertain because the values of θ_1 and θ_5 were falling and unsteady.

The conditions during the latter experiment for (*b*) were not satisfactory, but as no specific adequate reasons for the exceptional discrepancy of this result could be discovered, it has been included in the record.

Experiment No. 5. In this experiment the outer compartments (Nos. 1 and 5) were kept at a constant hot temperature, while the centre compartment (No. 3) was allowed to warm up under observation. It will be apparent that the theory of the method is the same as before.

The general agreement between the *cooling* and the *warming* methods will show that it is justifiable to assume that the boundary effects due to the frames and to external influences, could be considered as negligible within the numerical limits which are recorded. The results given in Table 7 and in Fig. 7 are for the ½ in. apparatus.

It should be noted that the difficulty of keeping θ_c constant makes the *warming up* tests much less reliable than those for *cooling down*. Not

TABLE 7. FOR HALF-INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_c-\theta_3)$ deg. cent.	$\log_{10}$ $(\theta_c-\theta_3)$
0	44.5	43.0	16.8	28.2	1.450
10	44.4	43.2	20.1	24.9	1.396
15	44.5	42.9	21.5	23.5	1.371
25	44.0	42.7	23.8	21.2	1.326
55	47.2	45.1	29.6	15.4	1.188
65	47.0	46.8	31.1	13.9	1.143
75	48.5	47.2	32.7	12.3	1.090
85	49.1	48.2	34.1	10.9	1.037

The average value of θ_c = 45.0 deg. cent. after subtracting 0.6 deg. to correct for the defects of slow circulation which lead to a cooling at the bottom of the compartments.

only was the temperature in the outer compartments variable instead of constant, but the circulation of hot water was slower and less regular, leading to differences in temperature between the top and bottom of the outer compartments which were not present in the *cooling down* tests. The accuracy of any one of these *warming up* experiments may be as poor as from 5 to 7 per cent.

The centre compartment (No. 3) was stirred for half a minute every 5 minutes.

The analysis of Fig. 7 shows that $B = 1.445$, $T = 94$, $L = 1.00$; hence: $k = 141.6 \times 1.845 \times 0.445/94 = 1.24$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

Experiment No. 6. The *warming up* method used in experiment No. 5 was repeated with the simultaneous use of both sets of frames giving the results shown in Table 8 and Fig. 8.

In the case of θ_c in the preceding table, 2.5 deg. for (*a*) and 2.2 deg. for (*b*) have been subtracted to correct for the effects of slow circulation on the lower parts of the compartments, and for the further cooling of the compartments as a whole, during the observations. The magni-

TABLE 8. (*a*) FOR THREE-QUARTER INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_c-\theta_3)$ deg. cent.	$\log_{10}$ $(\theta_c-\theta_3)$
0	40.1	39.0	17.3	18.1	1.258
5	38.8	37.8	18.0	17.4	1.240
10	38.0	37.0	18.7	16.7	1.223
20	37.6	37.1	20.0	15.4	1.188
25	38.0	37.1	20.6	14.8	1.170
30	37.9	37.0	21.2	14.2	1.152
35	37.4	38.0	21.9	13.5	1.130

Average for θ_c = 35.4 deg. cent.

TABLE 8. (*b*) FOR HALF-INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_c-\theta_3)$ deg. cent.	$\log_{10} (\theta_c-\theta_3)$
0	40.4	40.0	17.5	19.7	1.294
5	39.3	39.0	18.6	18.6	1.270
10	38.6	39.2	19.5	17.7	1.248
15	38.45	39.55	20.6	16.6	1.220
20	38.8	40.0	21.2	16.0	1.204
25	39.2	40.6	22.07	15.13	1.179
30	39.3	40.5	22.9	14.3	1.155
35	39.2	40.3	23.6	13.6	1.134
50	39.4	40.0	25.6	11.6	1.064
60	39.0	39.5	26.7	10.5	1.021
70	38.5	38.9	27.9	9.5	.978

Average for θ_c = 37.2 deg. cent.

tude of these corrections was determined by observations on the temperature gradient from top to bottom in the compartments (Nos. 1 and 5) and by the aid of the assumption that the graph should be linear, which is apparently justified by the other tests.

The analysis of Fig. 8 gives for (*a*) the following values: $B = 1.258$, $T = 98$, $L = 90$; hence since $l = 2.313$, we have: $k = 141.6 \times 2.313 \times 0.358/98 = 1.20$ B.t.u. per sq. ft. per hr. per deg. fahr. difference;

and for (*b*), $B = 1.294$, $T = 86$, $L = 0.90$ and $l = 1.845$; hence: $k = 141.6 \times 1.845 \times 0.394/86 = 1.20$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

Experiment No. 7. The last successful test of this series was made with the ¾ in. apparatus, along the lines of the last two experiments. The results are shown in Table 9 and Fig. 9.

It will be seen that in this case, $B = 1.317$, $T = 163$, $L = 0.70$, and therefore $k = 141.6 \times 2.313 \times .617/163 = 1.24$ B.t.u. per sq. ft. per hr. per deg. fahr. difference.

It may be concluded that the transmission of heat in each case is within 5 per cent of 1.25 B.t.u. per sq. ft. per hr. per deg. fahr. difference, and that there is less than a 5 per cent difference between the two sets of frames.

TABLE 9. FOR THREE-QUARTER INCH FRAMES

t min.	θ_1 deg. cent.	θ_5 deg. cent.	θ_3 deg. cent.	$(\theta_c-\theta_3)$ deg. cent.	$\log_{10} (\theta_c-\theta_3)$
0	40.9	41.2	17.0	20.9	1.320
5	40.8	41.5	18.1	19.8	1.297
10	40.9	41.7	19.0	18.9	1.276
20	40.0	40.9	20.7	17.2	1.236
35	39.0	40.1	22.5	15.4	1.188
45	39.1	40.0	23.8	14.1	1.149
55	40.0	40.7	25.0	12.9	1.111
65	40.3	41.6	26.0	11.9	1.076
75	41.4	42.5	27.0	10.9	1.037
85	41.6	42.8	28.1	9.8	.991
125	38.2	39.3	31.1	6.8	.832
135	39.2	40.6	31.3	6.6	.820

Average for θ_c = 37.9, corrected as in the last experiment.

TABLE 10. SUMMARY OF EXPERIMENTAL RESULTS AND CONCLUSIONS

Experiment	B.t.u. per sq. ft., per hr., per deg. fahr. difference — ¾ in. Frames	½ in. Frames
1	1.21	...
2	...	1.27
3	1.26	1.35
4	1.21	(1.45)
5	...	1.24
6	1.20	1.20
7	1.24	...
Average	1.22	1.27*

* Including the value 1.45, this mean value would be 1.30.

The arrangement of the apparatus in theses tests differs fundamentally from the practical case where our tanks of water are replaced by large bodies of air. As pointed out in a former section, the values obtained in these experiments should be about double the values for windows under ordinary conditions. If a 5 per cent difference is produced by the different characteristics of the insides only, of the double windows, the percentage difference when we take into account the addition of the outer air-glass contacts in each case will be reduced to about 2.5 per cent. It may be stated therefore that the two types of windows represented in these experiments transmit under conditions of usage approximately 0.6 to 0.7 B.t.u. per sq. ft. per hr. per deg. fahr. difference and that probably they differ between themselves by less than 2.5 per cent.

Although these experiments were tried only for differences in temperature up to 60 deg. fahr., it is clearly indicated by the linear character of the graphs that the same values of k will hold over a greatly extended range. From general considerations also it is to be expected that Newton's law of cooling as applicable in these tests, will be as valid for differences of 100 deg. fahr. as it is for those of 50 deg. fahr.

SUMMARY

1. The existing data available for calculating the transmission of heat through single-frame double windows are found to be conflicting and unreliable. Approximate preliminary calculations however, show (*a*) that the quantity of heat transmitted through these windows is of the order of half a B.t.u. per sq. ft. per hr. per deg. fahr. difference; (*b*) that the variation of this quantity with the thickness of the window is small; (*c*) that in the particular case of ½ in. and ¾ in. windows, the difference should be less than 5 per cent—probably much less.

2. A description is given of a simple piece of apparatus which was employed for making direct new determinations. The theory of the experiments and the methods of procedure are outlined. Several common difficulties in the measurement of heat transmission are eliminated.

3. In 10 determinations for ¾ in. and ½ in. single-frame double windows with their external faces in contact with water, it was found that with two exceptions all the determinations were within 5 per cent of 1.25 B.t.u. per sq. ft. per hr. per deg. fahr. difference, an average of 1.22 being obtained for ¾ in., and 1.27 for ½ in.

4. It is pointed out that when air replaces the water boundaries used in these tests, the transmission probably will be reduced by nearly 50 per cent. The percentage difference between the two windows will also be halved.

The final conclusion is that single-frame double windows of ½ in. and ¾ in. sizes, will transmit between 0.6 and 0.7 B.t.u. per sq. ft. per hr. per deg. fahr. difference; and that the percentage difference for the two sizes will be less than 2.5 per cent.

In presenting the paper upon these investigations, it is a pleasure to record the indebtedness of the author to Mr. F. W. Dakin of Sherbrooke, Quebec, and to Col. A. S. Eve, F. R. S., C. B. E., Director of the Physics Department, McGill University, for the provision of facilities. Many thanks are due also to Mr. J. A. Taylor, laboratory assistant, for able assistance in taking observations, and for overtime work.

DISCUSSION

L. A. Harding: I take issue with Professor Shaw in the statement that no accurate experiments have ever been conducted on the transmission of heat through glass. That is not the case at all, because we know of a number of the experiments made at different places that are practically in accord.

The tests refer to the conductivity of glass and not to the transmission with air contact, which are two entirely different propositions. He apparently made no determination on the combined radiation and convection factor. We have known for a number of years the conduction factor for glass. Perhaps the most accurate tests that have been run on the transmission of glass with air contact were made at the University of Illinois several years ago. Those tests were run in a very accurate manner on full-sized test specimens, and not by the water plate method or hot plate method.

James A. Donnelly: The general subject of transmission through building materials, glass and others, was one that I went over with Professor Allen in considerable detail in connection with the *National District Heating Association.* He was on a committee of which I was chairman, namely, the Committee on Research of the *National District Heating Association,* which has opportunity for some very broad research, and he outlined for some of our committee work the checking up of the losses through walls, windows, glass, and by infiltration, and the sum of all of the losses, yearly, daily, weekly and hourly as taken by meter readings and condensation. These data are available in the large plants, such as the Commonwealth Edison of Chicago, the Detroit Edison, and some Indianapolis and Dayton plants; there are loads of data that they have collected as to the transfer of heat through building materials.

To my mind the theory of transfer of heat and the practical research application today may be widely different things. Perhaps the district heating man, with the data available, can find more than any one else, not only on infiltration from the outside of the building to the inside, but the infiltration from floor to floor. Nothing has changed more in my practical experience of 35 years in heating than the filtration from floor to floor. Open elevator wells are no longer possible; they must be closed off by tight doors, so that we shut off a great deal of that infiltration. The stairways have become closed off through the *National Fire Protection Association's* efforts largely, so that we get very much lessened infiltration from floor to floor.

Mill construction buildings have a cold and heat leakage from floor to floor. Modern constructed buildings, with fire doors, closed stairways, and in many of the factories outside stairways, have made a change in the condition of buildings which can be detected in no other manner except by field tests of heating apparatus or condensation tests. I would like to welcome for the *National District Heating Association* as close cooperation as possible in a compilation of their records on condensation, such as would be suitable for the Research Laboratory and Director Scipio would be interested in.

A. NORMAN SHAW: Referring to my statement that the existing data for handling the problem (i. e., for the single-frame double window), appear to be conflicting and unreliable, L. A. Harding makes some criticisms which apparently do not refer to my work at all. He says (1) "I take issue with Professor Shaw in the statement that no accurate experiments have ever been conducted on the transmission of heat through glass"; (2) "His tests refer to the conductivity of glass and not to the transmission with air contact"; (3) "He apparently made no determination on the combined radiation and convection factor."

With reference to (1), it will be found that no part of my paper deals with the determination of the coefficient of conductivity for glass. Throughout the paper, it will be seen that the value for the conductivity of glass was assumed to be that which is at present usually accepted, and about which there is no particular controversy.

With reference to (2), it will be seen that the tests dealt with the total transmission through the combination of the following successive regions, glass, glass-air contact, air, air-glass contact, and glass, with the addition of the two water-glass contacts involved in the experimental procedure (which replaced only two of the air-glass contacts found in practical conditions).

With reference to (3) it should be clear that the combined effect of conduction, radiation and convection in the enclosed air space, was the very point examined.

No. 593

BRIQUETTED COAL FOR HOUSEHOLD FUEL

By J. H. Kennedy[1], Lykens, Pa.

Non-Member

THE briquetting of coal has been going on in Europe for over 50 years. At the present time it is a very large and extremely essential industry. Before the world war England was shipping to Italy alone over 3,000,000 tons of briquets a year. The briquetting industry has been developed more rapidly and successfully in Germany than in any other country.

The briquet industry[2] based on the brown coal deposits has all been developed within the past 20 years. It was stated that there were in the entire region of Horrem 15 years ago, only five factories in operation. In the vicinity of Horrem today there are 26 plants, the largest of which has 28 briquet presses and turns out about 1600 metric tons (1760 short tons) of briquets in 24 hours. About 20,000 metric tons (22,000 short tons) of brown coal are briquetted and shipped daily from this district during the season of active production.

The 26 briquetting plants have combined into a union or syndicate, with its chief selling agency in Cologne, the center of distribution for a population of more than 10,000,000 people. The capital of this syndicate is about 120,000,000 marks ($30,000,000) and the total output of the union is about 4,000,000 metric tons (4,400,000 short tons) or 400,000 carloads annually. This is more than one-quarter of the entire output of brown-coal briquets for all Germany during 1911, the last year for which statistics of production are available.

The industry seemed decidedly prosperous. The demand for the briquets was reported as steadily increasing and the returns on the annual investment in mines and factories was said to be very satisfactory. In the entire history of the region there have been no failures among the companies making brown-coal briquets.

Briquetting in the United States has been gradually growing, although a great many ventures into this field have failed. There are three main reasons for these failures, namely: *first,* a briquet sold in this country must be as good if not better than coal, and this is not the case in Europe; *second,* most of the machinery used has been (at least the types) brought from Europe to be used under entirely different circumstances without proper modifications to meet the new conditions; *third,* poor management and the lack of provision of the necessary capital for a period of experi-

[1] Superintendent, American Briquet Co., Lykens, Pa.

[2] Technical paper 55, Department of the Interior, Bureau of Mines, The Production and Use of Brown Coal in the Vicinity of Cologne, Germany, by Charles A. Davis.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1921.

ment, which almost invariably is necessary. Nevertheless, a steady increase is noted and below is given the tonnage produced in the United States from 1907 to 1919[1]:

Year	*Quantity Net Tons*	*Year*	*Quantity Net Tons*
1907	66,524	1914	250,635
1908	90,358	1915	221,537
1909	138,661	1916	295,155
1910	No figures given	1917	406,856
1911	218,443	1918	477,235
1912	220,064	1919	295,734
1913	181,859		

It is rather difficult to give a discussion on briquetting in general at its present stage of development in this country, because of the lack of definite specific knowledge in relation to it. The field is large and undeveloped, at present; but the time is rapidly approaching when it will become an absolute necessity even in this country as it is at present in Europe.

As the briquets made from anthracite culm are the best for household consumption we will continue further discussion to the briquetting of anthracite culm and particularly by means of the C. E. Hite binder process. Furthermore, before entering into the manufacturing process a word should be said as to the essential features of a good briquet. A briquet must stand tests to prove that it is: 1. Hard and tough, so as to withstand shipment; 2. Able to withstand the attack of the elements (especially water and heat); 3. Able to retain its shape while burning; 4. Not likely to produce any smoke or undesirable odor while burning. The following materials have been tried as binders under the supervision of the Bureau of Mines: rosin, pine tar, sulphite liquor, starch, tars, magnesia, magnesia cement, plaster of Paris, Portland cement, water glass, and slag cement. These have all been used with varying degrees of success.

COMPOSITION OF C. E. HITE BINDER

The C. E. Hite binder, now owned and used by the American Briquet Co. at their plant at Lykens, Pa., is an emulsion of water, starch and asphaltum. This is commonly called the water binder. It is made with the following constituent parts: 200 gal. of water, 140 lb. of Globe pearl starch; 35 gal. of asphaltum. The preparation of the binder is as follows: We take 25 gal. of water and mix with it 140 lb. of Globe pearl starch. Then we heat 175 gal. of water to the boiling point. The boiling water is then dropped into a mixer and agitated while we add to it the mixture of cold water and starch. This forms a thick starch paste. As soon as the starch paste is thickened, we add 35 gal. of asphaltum at 220 deg. fahr. The asphaltum must be added very slowly at first, or we will not obtain the proper emulsion. After the emulsification is complete, the mixture is dropped into the reserve binder tank for storage, and is ready to be used for the manufacturing of briquets. The mixing is entirely completed in about 4 min. This binder is then composed of 79 per cent water, 6 per cent starch and 15 per cent asphaltum—by weight.

[1] Fuel Briquetting in 1919, 11.2—Department of the Interior, by F. G. Tryon, Mineral Resources of the United States, 1919, Part 2 (pp. 33-36).

In preparing the culm for its mixture with the binder we must first remove a great deal of the moisture. It will generally be necessary to either wash or crush the culm first; but because of the excellent condition of the bank at Lykens we do not have to do this. In tests we have made of this culm, as it is in the bank, we have not as yet found over 11 per cent ash at any point. Nevertheless, we must dry it. This is done, primarily to secure the effect of capillary attraction in the mixing of the binder with the culm. The moisture of the culm as it comes from the bank varies from 10 to 16 per cent, and we must dry it down to 1½ to 2 per cent. This is done by means of a single steel rotary drier, 50 ft. long and 7½ ft. in diameter. At one end of the drier there is a furnace 7 by 7 ft. Air is pulled through this drier by means of a fan, passing the air and hot gases directly through the drier, in direct contact with the culm, thereby removing the moisture. One can readily appreciate that as the coal is dried a great quantity of dust is carried out with the exhaust air. This feature has given us considerable trouble; and even to the extent that the small town of Wiconisco obtained an injunction to postpone our operation until we eliminated the nuisance. We had been experimenting for some eight months before we finally eliminated it. We tried a cyclone dust collector, a precipitation chamber, passing the air through water sprays, and even through water. None of these arrangements eliminated the nuisance, although a certain proportion was collected. This very fine dust carried as far as 2 miles from the plant.

We are handling through this drier approximately 12,000 cu. ft. of air per min. At the suggestion of Charles Lutz, now experimenting with the powdered fuel for the M. A. Hanna Co., a collector was built, so that the speed of the air at the entrance to the collector was 4000 ft. per min. The proportions of the collector are those used by the York Heating & Ventilating Corp., York, Pa., for their standard dust collector, with the exception of the tubular cone. This we lengthened so as to increase the centrifugal effect in the cone before the air had an opportunity to exhaust. We then entered a 1½ in. water line at the point the air enters, and turned it the full circumference of the collector. This pipe has ⅛ in. hole every 6 in. and the result is we keep the sides of the collector always wet with water. The dust is thrown into the water; and, although it does not seem to mix with the water, it is washed out and our fan exhaust is apparently steam.

MIXING BINDER AND CULM

We mix the binder with the culm by means of a split-screw conveyor and convey it to the press. We are using the Belgian-roll type press. The press is simply two 40 in. diameter rolls running in opposite directions. In each roll, small pockets are cut so as to meet up exactly as the rolls pass each other. The material is then fed into the pockets and squeezed into shape between the rolls. Approximately 10,000 lb. per sq. in. pressure is required for a good briquet. This varies unless a constant feed is maintained as the face of the rolls must be kept 1/32 in. apart. From the press the briquets are conveyed to a drier.

In mixing the dried culm (approximately 2 per cent moisture) with the binder we use 92 per cent culm and 8 per cent binder. These are the percentages which obtain in the briquet as it leaves the press. As has been said above, the binder is composed of 79 per cent water, 6 per cent starch and 15 per cent asphaltum; and 8 per cent of the briquet is this mixture, as it leaves the press. In the final drying we again reduce the moisture to 2 per cent. This means that we remove all the water which was added to the culm by means of the injection of the binder. The resultant is therefore: 96.21 per cent culm; 2 per cent moisture; 0.50 per cent starch; 1.28 per cent asphaltum. The moisture content is the most variable. This varies from 1 to 2.5 per cent. One analysis which I will give later shows 1.17 per cent.

The function of the final drying is twofold: *First,* we have in the briquet too much moisture—this must be removed not only for burning, but to allow the starch to harden; *second,* in drying we must raise the temperature of the air to 325-375 deg. fahr. in order to dextranize the starch. This adds to the starch not only hardness but toughness. The drying is accomplished by forcing hot air through the briquets as they are passed through the drier on a wire apron conveyor.

In the use of anthracite briquets as a household fuel there are three main advantages for the right kind of briquets:

1. Very rapid and complete combustion, due to the even size and shape of the briquet;

2. The lack of clinkers in the fire. We burned briquets all last winter (1919-1920) in our home and did not have a clinker. There are two reasons for this: *a.* the culm is the fines from mining and the pure coal pulverizes more rapidly than slate or rock; *b.* most of the culm has come through a washery.

3. A briquet fire requires much less care and will keep longer when properly banked than the sized coal from which the culm is obtained.

BRIQUETTING PROBLEMS IN THE UNITED STATES

Coal in the briquetted form has, as a rule, been seriously condemned in the United States. In most every case this condemnation has been warranted. The briquetting problems in the United States are much more complicated than in Europe, because here a fuel must be made that is as good or better than the coal from which it is made, while in Europe the only requirement is that it must burn. A great many binders used, require anywhere from 6 to 12 per cent of a substance that produces an undesirable smoke and odor in burning. Others are not water or heat proof. Nevertheless, with the C. E. Hite binder over 98 per cent of the finished product is culm and moisture. Below is an analysis of the briquets that are now being made. This analysis was made by the New England Fuel & Supply Co., Boston, Mass.:

	Per Cent
Moisture	1.16
Volatile matter	13.10
Carbon	75.00
Ash	10.10
Sulphur	.64
B.t.u.	14,090
Crushing strength	250 lb.

In using briquets as a household fuel one must learn not to leave the draft on the fire too long. As mentioned above that the combustion is rapid and complete, because of the even size of the fuel. The result is that when anyone is first using briquets, he will either burn the lids on the stove; or when preparing for the night, will not check the draft enough and the fire will burn out. This must be learned by experience. On a windy night it is necessary to cover the fire with a thin layer of ashes before putting on the briquets for the night so as to check the draft. The fire looks as though it could not possibly last, but it does.

The ash produced by briquets is very fine—almost a powder, and it must be handled carefully for it is easy to raise a dust through the house. This is the one undesirable feature that has been found so far, but with a little care this is easily overcome, and the advantages far outweigh this one disadvantage.

DISCUSSION

R. P. Bolton: What is the shape of the briquets?

The Author: The briquets are pillow-shaped. They weigh just about two and one-half ounces and they look like a very small pillow, being made by means of a Belgian press. The pockets are cut on the rolls of the press, and as the rolls go around and meet, the material is fed between and put under a pressure of approximately 10,000 lb. per sq. in.

R. P. Bolton: Is the size of the briquet roughly about the size of stove coal?

The Author: Yes, about the size of stove coal.

J. G. Dudley: Some 20 years ago I made a test of briquets which were submitted to me, hence I believe you may be interested to know of my recent experience of some three or four months' operation with modern briquets. In September the coal dealers in Philadelphia advised me that there was no coal obtainable which would burn on my grate save pea coal, which would go through it, and so was disadvantageous, because it could not be burned without great effort.

I finally bought six tons of briquets, about the size of golf balls. I went back of the dealer to the manufacturer's technical authority to get instructions, because it was a most interesting combustion experiment, and my comfort was predicated on it.

It was represented that the ash was not greater than 18 per cent and that the binder was a heavy oil. It ignites and burns readily, it disintegrates readily, and it clogs up readily. However, it gives off heat readily and by great care one may get satisfactory results with an ordinary heating system, such as mine which happens to be a vapor system of an excellent character. The results upon coming of cold weather, were so sensitive that I had to give it much attention. The ash fills up into a very heavy bed and clogs like sand, so that the air does not get through it. If I break it up frequently to let the air get through it, I get just what the speaker has told us we would get; that is, good results.

The discussion this afternoon on briquetting fuel it seems to me missed certain features that the engineers in heating, and combustion efficiency engineers especially, should be analyzing. In other words the discussions seemed to be predicated on the assumption that the types of apparatus and the methods of combustion and the methods of use for warming our houses, as conventionally employed, are going to continue. I do not consider that this is so at all. As a matter of fact, I am quite positive that the reverse is true. I have knowledge of matters impending right at this moment which are apparently going to replace many of our present methods.

The question of briquetting is undoubtedly what the speaker has said it is possible to do with coal. But why briquet it? It is because we cannot get the true anthracite in the proper marketable condition to burn in our furnaces. Only this day I learn that Senator Calder seems to be taking up the cudgels for the engineering profession as well as the householders on this score of the proper rating of coal sizes.

The need for the briquetting of coal is a question of size; since the rest of the coal dealers do not stick to the standard sizes that are accepted by the engineering profession, with the result that it makes trouble all along the line.

No. 594

ECONOMIZERS

By W. F. Wurster[1], Philadelphia, Pa.
Non-Member

THE increased cost of operating steam generating plants has caused the owners, operators and designers to go into the subject of economy more thoroughly than heretofore, in order to obtain the maximum amount of steam per pound of fuel burned. One method for lowering the cost of steam is by the use of an economizer in conjunction with the boilers.

An economizer consists of a number of cast-iron tubes located in the gas passage from the boiler and around which tubes the gases of combustion pass on their way from the boiler to the stack. The feed water, on its way to the boiler, passes through the tubes of the economizer, absorbing a large proportion of the heat which would ordinarily be wasted.

The economizer was first used in 1845 by Edward Green. It consisted of a circular group of 30 cast-iron tubes, 4 in. in diameter by 9 ft. in length, connected at the top and bottom to hemispherical chambers, the cold water being introduced into the lower one, and after rising through the tubes, passing off to the boiler through a branch at the top. This apparatus heated the water to the boiling point and at times formed steam which was allowed to escape through a safety valve at the top.

In a few weeks' time it was observed that the temperature of the water passing from the economizer gradually became lower, the saving in coal became correspondingly less and the draft was impaired. The brick work enclosing the machine was removed and it was found that the gas-passage was choked up with soot, and that the tubes were partially insulated by the deposit of this soot upon them. The above observation led Mr. Green to design mechanical scrapers as an automatic method for preventing the accumulation of soot upon the tubes.

The modern economizer is built in sections each of which is composed of a top and bottom header with the necessary tubes. The top headers are machined on the sides so as to fit closely together, thus preventing the escape of gases from the economizer or the infiltration of air. Adjacent headers have a semi-circular groove cast vertically on opposite sides and when the sections are placed in position, these grooves come together, forming a hole for the passage of the scraper chain.

The bottom header is made ovoid in shape in order to leave an extra large space between the sides of adjacent headers. This prevents the

[1] District Manager, Green Fuel Economizer Co., Philadelphia, Pa.

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banking or bridging of soot across from one header to another and also allows room for the insertion of a man's arm to adjust or replace scrapers should this become necessary. One end of the header is extended horizontally to pass beyond the side wall of the economizer chamber. This extension is flanged for connection to the bottom branch pipe.

The headers previously described are provided with tapered sockets to receive the ends of the tubes. One method is to have both ends of the tube taper in opposite directions. In this case the tubes are pressed simultaneously into the top and bottom headers. Another method is to have the ends of the tubes tapered in the same direction. In this case the tubes are inserted through the top header.

Sections are built with 4 to 12 tubes each. These tubes vary from 6 to 12 ft. in length and are 4-9/16 in. in diameter. The number of sections

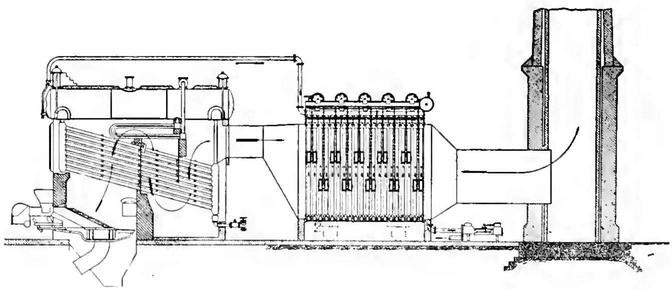


FIG. 1. CROSS-SECTION OF A TYPICAL BOILER PLANT SHOWING INSTALLATION OF AN ECONOMIZER

in an economizer is a multiple of four which is due to the fact that the pulley on the scraper driving mechanism operates the scrapers for four sections.

The bottom headers are connected to a bottom branch pipe. This pipe is flanged on one end to receive the feed line and on the other end for a blow-off valve. The top headers are also connected to a top branch pipe for collecting the heated water. One end of the branch pipe is flanged for connection to boiler feed line. The other end is blanked. Provision is made for a relief valve on the top branch pipe.

The exterior of each tube is kept free from soot by means of scrapers which travel continuously up and down the outside of the tubes at a slow speed. The soot falls into the soot pit beneath the economizer, whence it can be removed at regular intervals.

The lifting bar carries the scrapers for two adjacent sections and the guard holds the scrapers in place on the downward stroke. The lifting bar, guard and scrapers for two sections are hung from one end of a chain which passes up through the chain hole between the top headers, over a chain pulley carried on a heavy cast-iron frame erected on the top of the economizer. The other end of the chain passes through the next chain hole and is connected to the lifting bar, guard and scrapers for the next two sections. In this way the one set of scrapers is balanced by the other. This arrangement reduces the amount of power required to operate the scraper mechanism. The power required for operating the scrapers is about 1 hp. for 125 tubes.

The mechanism for reversing the direction in which the scrapers travel consists of a shaft running the length of the economizer supported by the gear frame. Worms on this shaft engage with worm wheels which are bolted to pulleys, over which the chains supporting the scraper bars and guards are run. The reversing is accomplished by a clutch-and-pivoted reversing lever which has a rolling weight to insure quick and positive engagement. A cross shaft is mounted in back of the reversing breast and driven from the first worm wheel. A worm on this shaft drives, through an idler, a wheel which is mounted on the top of the frame and in back of the pivoted reversing lever. Two adjustable fingers on this wheel engage the reversing lever and force it first to one side and then to the other, as the wheel carrying the fingers is revolved

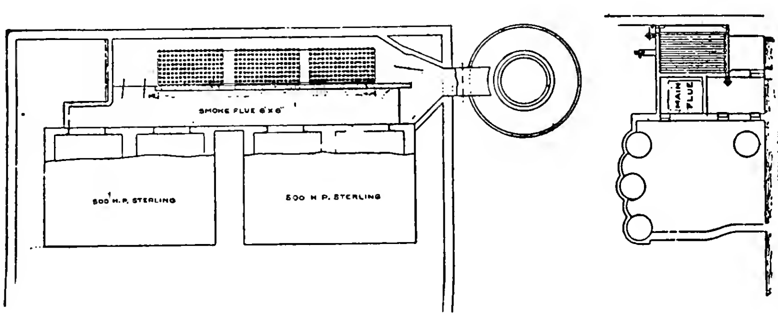


FIG. 2. LAYOUT OF A TYPICAL BOILER PLANT SHOWING INSTALLATION OF ECONOMIZERS AT REAR OF SETTINGS

through an arc, first in one direction, and then in the other direction. The change in the position of the reversing lever causes the clutch to engage alternately the right and left hand beveled pinions, transmitting motion to the longitudinal shaft.

The sectional covers consist of two steel plates with about 2 in. of asbestos or other insulating material between. This method of enclosing one or both sides of the economizer is now preferred to the older method of bricking in the machine as it permits easy access for inspection and cleaning.

In determining the saving which can be made by an economizer, various factors are taken into consideration. The most important of these factors are the temperature and quantity of gases available, and the initial temperature and quantity of the feed water which is to be heated.

In existing plants the height of the stack becomes a factor to which due consideration must be given. Many plants have stacks which were figured to take care of 120 lb. of gas per hp. per hour, and in such cases the economizer can be installed with very little reduction in the draft. In modern plants where high ratings are developed at high pressures and with superheat, induced draft is used in preference to high stacks.

The gas temperature, from boilers run at rating, varies from about 470

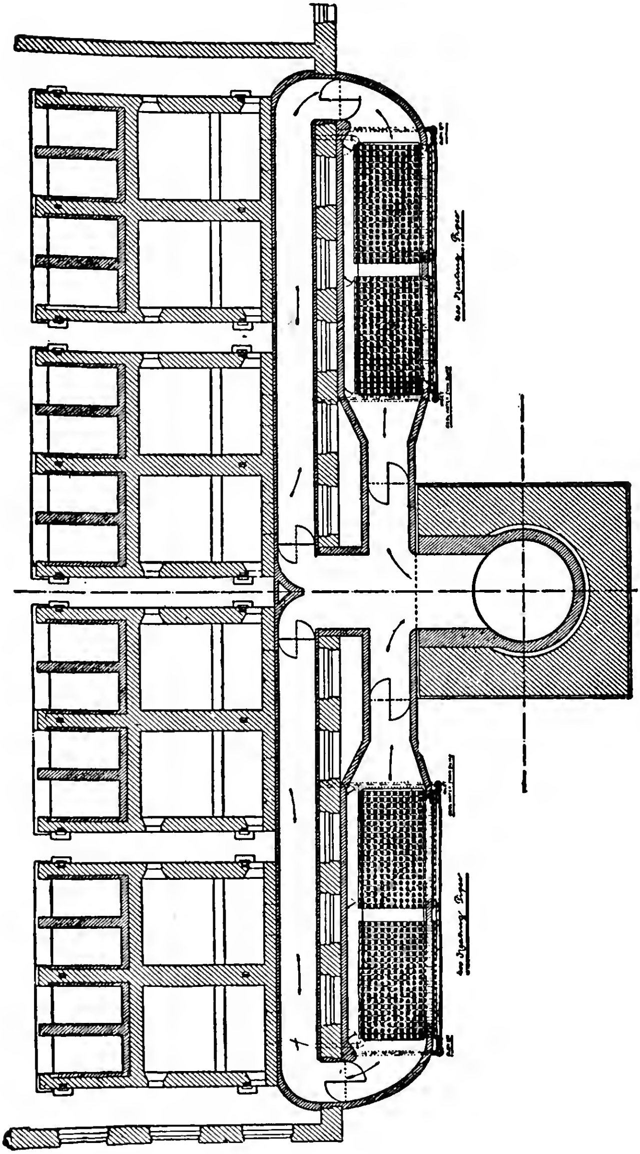


FIG. 3. LAYOUT OF LARGE BOILER HOUSE WITH ECONOMIZERS LOCATED OUTSIDE, SHOWING FLUE CONNECTIONS AND METHODS OF BY-PASSING

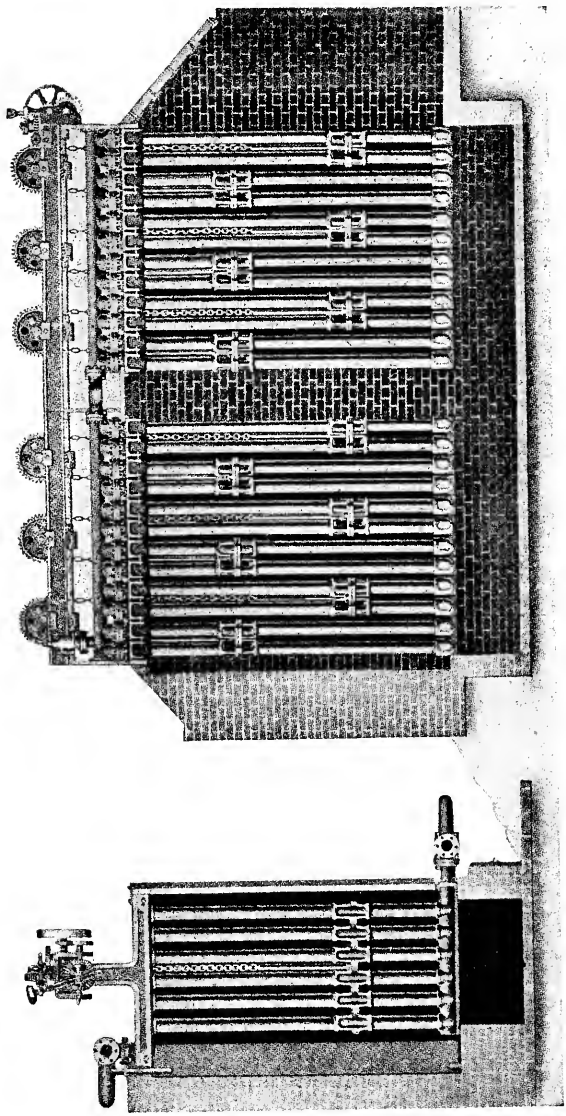
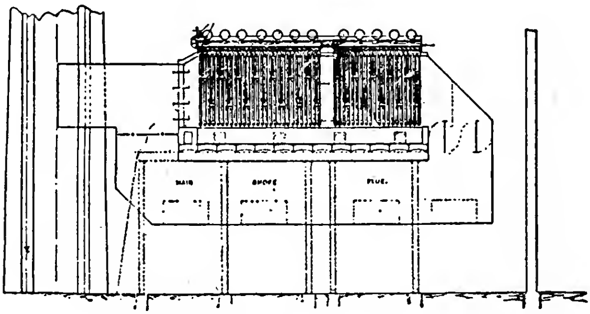


FIG. 4. SECTIONAL VIEWS OF A TYPICAL ECONOMIZER INSTALLATION

to 550 deg. At 150 per cent rating the temperature varies from about 525 to 600 deg., and at 200 per cent rating from about 575 to 675 deg.

The quantity of gas available depends upon the class of fuel and the method of firing the same. In underfeed stoker practice about 60 lb. of gas per hp. per hour are available for heating the feed water; with good hand firing or with natural draft stokers about 80 lb. of gas per hp. per hour are obtained, and with ordinary hand firing from 90 to 100 lb.



Longitudinal Elevation

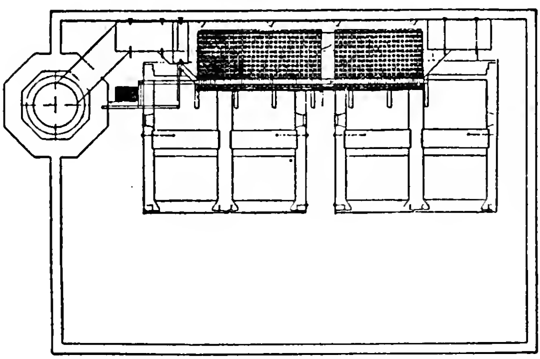


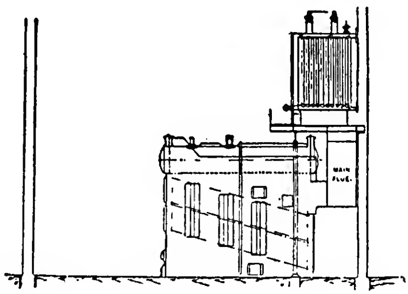
FIG. 5. LAYOUT OF TYPICAL BOILER PLANT SHOWING INSTALLATION OF ECONOMIZER ABOVE BOILER SETTINGS

per hp. per hour are obtained. In an oil fired plant the quantity of gas per hour per hp. is about 50 lb.

The temperature of the feed water ordinarily ranges from 100 to 210 deg. The quantity of water per hour per b.hp. depends upon the initial feed temperature, steam pressure, and super heat but is usually close to 30 lb.

In an industrial plant containing five 300 hp. boilers, one of which is held as a spare, one economizer is capable of handling the feed water. In this case the steam pressure is 150 lb., gas temperature 525 deg., 80 lb. gas per b.hp. available for heating the feed water which has an initial temperature of 120 deg. A machine of 320—10 by 10s gives a rise of 97 deg. in temperature or a saving of 8.7 per cent of the fuel burned. This plant operates 10 hr. per day and burns 24 tons of coal at a cost of $151.70. The economizer saves 8.7 per cent of this amount, or $13.20 per day, and in a year of 310 days the machine saves $4,100. This machine installed cost $12,000 and with due allowance for interest, depreciation, maintenance and repairs shows a return of about 20 per cent on the investment.

Let us assume a modern plant containing six 1000 hp. boilers, equipped with underfeed stokers operating normally at 160 per cent rating with a peak load of 200 per cent of rating; 60 lb. of gas per hour per b.hp. available at a temperature of 560 deg. at 160 per cent rating, and 600 deg. at 200 per cent rating; steam pressure, 200 lb., superheat, 100 deg., initial feed temperature 175 deg.



Cross Section

FIG. 6. CROSS-SECTION OF TYPICAL BOILER PLANT SHOWING INSTALLATION OF ECONOMIZER ABOVE BOILER SETTINGS

An economizer of 384 tubes installed in connection with each boiler would raise the feed water 80 deg. to a final temperature of 255 deg., effecting a saving of 7.2 per cent. Running 24 hr. a day the yearly coal consumption would be approximately 28,000 tons per year, which at $5.00 a ton amounts to $140,000, 7.2 per cent of which is slightly over $10,000. The economizer and its appurtenances could be installed for a sum not exceeding $15,000. Allowing 15 per cent of the cost of installation for interest, depreciation, maintenance and repairs and subtracting this amount from the yearly saving, would leave $7750 net saving by the use of the economizer, which corresponds to 51 per cent on the investment.

The cost of generating steam in a small plant is naturally greater than in a large plant, due to less efficiently operated boilers and lack of refinements. An economizer in a small plant operating from 300-500 hp. shows up as well and in some cases better than a machine in a large plant. Economizers have been installed in plants operating less than 300 hp., but in these cases special machines were built to handle the smaller quantities of gas.

ADVANTAGES OF ECONOMIZER

In addition to the saving made by an economizer there are further advantages to be obtained: *First,* an increase in the capacity of the plant;

Second, there is always a large supply of heated feed water on hand which is a great help in handling fluctuating and peak loads; *Third,* the boiler is protected from cold feed water which reduces expansion and contraction strains. Another advantage is the fact that a large portion of the impurities found in the water used for boiler feeding is precipitated in the economizer because of the slow motion of the water in the tubes. These precipitates can be blown off or washed out as a sludge; whereas if they have been allowed to form in the boiler they would have become baked on the tubes as a hard scale which is difficult to remove. The economizer can usually be cleaned by means of a stream of water from a hose connected to the service line.

Economizers are used for a number of purposes besides heating boiler feed water. Among these are heating water for hot-water service in hotels, for wash water in laundries, for process work in industrial plants. They are used as brine heaters in salt plants; also as preheaters for oil in oil refineries.

Many economizers are operating on the waste gases from cement plants, oil stills and other industrial operations where the waste gases are discharged at a temperature of over 350 deg.

The increase in the cost of fuel, both coal and oil, indicates that the demand for economizers will be greater than heretofore.

No. 595

PROPER SELECTION OF HOT-WATER HEATING DEVICES FOR DOMESTIC SERVICE

By Albert Buenger, St. Paul, Minn.

Member

THE heating and ventilating engineer finds it necessary from time to time to select hot-water generators, design and size hot-water supply piping, select circulating pumps and the like, but I do not believe there is any available authority which gives a complete and feasible method for doing this. Most of the information available today is in the form of magazine articles, and a large amount of real valuable information that the manufacturers of various heating devices and specialties have developed. As I have on different occasions had to select apparatus and equipment for hot-water service, I have gathered information from a number of different sources and have compiled it in usable form for application to the different problems which present themselves. Much of the data will be recognized by the members as having appeared in the columns of the Journal during the past few years or in the trade publications.

The methods of computation in this practice are not based on such accurate scientific information as most of our heating and ventilation computations on account of the personal element which enters into the determination of the probable demand; it is the determination of the probable demand which forms the basis of all computations. After this is once known, the selection of the heaters, sizing of pipes, selection of pumps and the like is merely a matter of simple arithmetic.

It is true that the number and kind of fixtures are usually selected before the engineer approaches the problem, and it is a comparatively simple matter to determine the amount of water flowing per minute through the various types of bibbs and faucets at a certain definte pressure; but the pressure of the water as well as the incoming temperature are variable which would alter the demand considerably. Even a variation in temperature and pressure can be taken care of in the computations by assuming the worst conditions as a basis; but the item which really complicates the problem and makes an accurate scientific solution difficult, if not impossible, is the personal element which enters. Under certain conditions, this item can also be estimated with fair accuracy. Take for example, an apartment house or home; here the engineer can

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closely approximate the maximum probable demand and at what times of the day this demand will occur. In other installations, such as hotels, hospitals, clubs and similar buildings, the demand is extremely variable and cannot be estimated with any semblance of accuracy. As there are so many variables which enter into the problem, it resolves itself eventually into the application of good common sense and judgment coupled with what meager facts we have at hand.

TABLE 1. FLOW IN GALLONS PER MINUTE DELIVERED BY ORDINARY PLUMBING FIXTURES

Fixtures	Fair Flow	Good Flow	Excellent Flow
Kitchen-Sink Bibbs	2	4	6
Pantry Sink—High Goose-Neck Bibbs	2	2	3
Pantry Sink—Large Plain Bibbs	4	6	8
Vegetable-Sink Bibbs	2	4	6
Laundry—Tray Bibbs	4	6	8
Slop-Sink Bibbs	3	4	6
Lavatory-Basin Bibbs	2	3	4
Bath-Tub Bibbs	3	4	6
Shampoo Spray	½	1	2
Liver Spray	1	2	3
Shower Baths			
5 in. Rain Heads	2	3	4
6½ in. Rain Heads	2	3	5
8 in. Rain Heads	4	6	8
8 in. Tubular Heads	6	8	10
Needle Baths	20	30	40
Manicure Table	1	1½	2

The first step in determining the probable demand is to ascertain the number and type of fixtures which are to be installed. From Table 1 the flow per minute from each fixture can be determined. This table was compiled by actual tests on a water pressure of 30 lb. per sq. in. It is intended to set forth what is considered only a proper flow from the fixtures. It does not give the largest flow in any case; that is governed by the water pressure. Differences will be found among similar fixtures made by different manufacturers. In explanation of the three rates of flow listed, it should be noted that by *Fair Flow* is meant a stream just large enough to render what might be called good service; by *Good Flow* is meant a stream which in most households would be entirely satisfactory, and by *Excellent Flow* is meant a flow which if increased to any great extent would cause annoyance by splashing and noise.

Table 1 or a similar one, represents about the limit of our definite technical information which would be of value in determining the demand. But even here the values given in the table are variable, depending on water pressure, variation in the construction, and design of the different bibbs or faucets, and again the personal element dictating just what is considered good flow. From this point all of the rest of the determination is based on more or less indefinite factors. First, it is necessary to determine the number of usings per hour and the length of time each fixture is used during the period of greatest demand for hot water. This will give us the rate of flow during the peak load. Then, it will be necessary to determine, if possible, the period over which this peak load extends and at what intervals it occurs. It will also be necessary to estimate the hourly demand during the off-peak periods.

For each different installation or rather type of installation, new values must be determined in order to arrive at a result which will enable the designers to select the proper apparatus. In some of the simpler installations, such as apartment houses or institutions where more or less regularity is maintained, the matter of gathering this information is not so difficult. However, as soon as the installation becomes more complex the determination of these facts becomes extremely difficult, if not impossible. After the demand has been determined in this manner, it will be necessary to select a heater whose total or potential capacity would be sufficient to supply hot water during the entire peak load, and whose heating capacity would be large enough to supply hot water during the off-peak load in addition to heating the water in the storage tank.

TABLE FOR HEATER SELECTION

In order to facilitate the selection of heaters, tables have been prepared and are in use by different manufacturers for this purpose. Table 2 is a combination of those used by two well-known manufacturers. The upper part of the table takes into account the number of usings per hour of each different fixture in the various installations. The next row of percentages shows the utilization factor while the last row of percentages indicates the probable distribution of the peak load over a certain length of time, but is actually expressed in the capacity of storage tank in terms of the total maximum heating capacity.

It will be noted in the above table that 150 deg. fahr. is the standard temperature required at the fixture. Most of the heaters are rated for 180 deg. fahr. final temperature. This would take into account, loss in temperature from the tank to the fixture.

The table also includes a rating for apartment houses based on the number of families in the building. The per capita method of determining the demand is not very satisfactory as the demand varies with the size of the institution and the ultimate water consumption is actually dependent on the number of fixtures installed. For hospitals, for example, the demand has been estimated from 100 to 150 gal. per day per person. This figure is for the entire demand and includes hot and cold water supply and is based on both patients and employees. This ratio does not apply to all cases. An investigation of the various hospitals in the Twin Cities showed that the consumption was from 100 to over 300 gal. per day per capita.

In addition to the hot water demand of ordinary plumbing fixtures, there are very often other hot-water-using apparatus such as laundry equipment, kitchen equipment, or swimming pools. Laundry equipment is a class of equipment which uses a large amount of hot water. No rule can be laid down for the determination of the hot water demand in a laundry on account of the wide variation in the methods on which the

TABLE 2. HOT-WATER-FIXTURE CAPACITIES FOR VARIOUS TYPES OF BUILDINGS

Gallons of water per Hour per fixture, figured at a final temperature of 150 deg. fahr.

Fixtures	Apt. House	Club	Gym.	Hosp.	Hotel	Indust. Plant	Laundry	Office Bldg.	Public Bath	Private Res.	School	Y. M. C. A.
Private Lavatory	3	3	3	3	3	3	3	3	3	3	3	3
Public Lavatory	5	8	10	8	10	15	10	8	15		18	10
Bath Tubs	15	15	30	15	15	30			45	15		30
Dish Washers	15	30		30	30	30				15	15	30
Foot Basins	3	3	12	3	3	12				3	3	12
Kitchen Sink	10	20		20	20	20				10	10	20
Laundry Stat. Tubs	25	35		35	35		42			25		35
Laundry Rev. Tubs......	75	75		100	150		75 to 100		100	75		100
Pantry Sinks	10	20		20	20					10	20	20
Showers	100	200	300	100	100	300			300	100	300	300
Slop Sinks	20	20		20	30	20	10	15	15	15	20	20
Dish Washers	200 gal. per hr. at 180 deg. for serving capacity of 500 people.											
Total Water for all Fixtures likely to be drawn at one time. Per Cent..............	35	60	80	75	60	90	100	20	100	50	25	75
Storage cap. in per cent of max. heating cap....	35	40	40	45	45	50	50	40	50	30	40	50

APARTMENTS—ONE KITCHEN, ONE BATH

No. of Families...........	Up to 25	25-50	50-75	75-100	Over 100
Gals. per Hour..........	35	30	25	20	15

SWIMMING POOLS

Usually figure to fill pool in 24 hr. Also figure on refiltering all water in pool every 24 hr. 50,000 gal. pool is sufficient for 100 persons.

laundries are operated. The manufacturers of laundry equipment have information as to the amount of hot water required for each type of equipment, and the service required in the laundry must be determined. By proper operation, the water consumption in a laundry can be cut down a large extent. I have in mind a case which recently came to my attention in which the water consumption of a laundry was cut down from 40,000 gal. per day to 25,000 gal. per day.

The kitchen equipment demand can be estimated in a manner similar to the laundry equipment.

In estimating the demand for swimming pools, it is usually figured to fill pool in 24 hours and also to refilter all water in pool once every 24 hours. A pool for 100 persons has a capacity of about 50,000 gal.

After the demand on the heater has been determined, the problem is narrowed down a large extent to a selection of the proper apparatus which will deliver the amount of hot water required at the various fixtures.

Water heaters can be classified into two large classes: 1. Instantaneous or flash-type heaters. 2. Storage-type heaters. Instantaneous heaters are limited in the larger sizes (not gas heaters) to installations where the demand is continuous, as is the case in swimming pools or in industrial installations. Storage-type heaters can and should be used on all installations where the demand is variable. An instantaneous heater installed where the demand is variable would need to be of sufficient capacity to supply the demand at maximum conditions. At these times, the steam required to heat a large amount of water would be excessive. Another objection which arises is the fact that after the water demand ceases, the heater is filled with steam, causing the water in the comparatively small space in the tubes to overheat and generate steam, which when drawn shortly after on a similar demand might lead to disastrous results. Then the third objection which must be considered is the fact that for maximum demand, the steam supply and steam regulator must be of such size as to furnish steam to the heater without undue pressure drop. This would call for a size of steam supply connection which under off-peak condition would cause the regulator valve to open just slightly and the steam would wire-draw on the seats of the regulator and in time make regulation impossible. This last objection, however, can be overcome by proper installation.

STORAGE TYPE HEATERS

The storage-type of heater is preferable in most installations where the demand is spasmodic or variable. In selecting a storage type of heater, the proper relation between the storage and heating capacities of the heater are of importance. As has been stated above, the potential capacity, that is, the storage capacity plus the heating capacity of the heater, must be of such size as to supply the peak load during the entire period of the peak. The heating capacity must be of such size as to heat the water during the off-peak period in addition to heating the water in the tank for storage for the peak load. It can readily be seen that this type of heater will eliminate the objections of the instantaneous heaters for this type of installation. The steam requirements at a single instant would not be excessive but would be distributed over a longer period with a smaller maximum demand; also the storage capacity of the heater eliminates the possibility of scalding due to excessive temperatures.

Another classification of water heaters is according to the heating medium. There are four different mediums which are more or less popular, as follows: coal, steam, gas, and electricity.

Coal heaters are not usually installed for supplying large amounts of hot water but are ordinarily used for stand-by service in conjunction with a steam heater or in places where a steam supply is not available in summer and the coal heater is used exclusively as a summer heater.

The engineer here has his choice between the steel and cast-iron heaters. The various manufacturers publish capacities and ratings of these heaters. The capacity of any particular heater is limited to its coal-burning capacity. Then the shorter the firing period, the larger the heating capacity, as the coal in the firepot will be burnt at a faster rate. In selecting a cast-iron heater, it is well to bear this fact in mind as most of these heaters are based on a firing period of from 5 to 8 hours. For this reason, a much smaller heater can be selected by shortening the firing period. Another fact to be remembered in selecting a cast-iron boiler is that the working pressure of most heaters is limited to about 40 lb., and they will not stand 100 per cent overload. It has been my experience that they will crack at from 80 to 100 lb. pressure. Steel heaters are usually designed for 100 lb. working pressure.

A type of heater which is only used in small installations is the range boiler and water back. The water back as usually installed in kitchen ranges is made of cast iron, and the ordinary water back has a capacity of 25 to 35 gal. of water per hour per sq. ft. of surface exposed to the fire. The average-size water back has about 110 sq. in. of exposed surface and will heat about 17 to 27 gal. per hour. The size of range boiler to install in a residence is variable, depending upon the number of bath rooms in the house. It is well to have a range boiler of ample capacity as the water backs are usually rather small.

STEAM HEATER TYPES

Probably the most widely-used heater for the larger installations is the steam heater. The steam heaters can again be divided into two general groups—the water-tube type and the steam-tube type. Practically all of the storage-tank type of heaters are of the steam-tube type, and few, if any, of the water-tube heaters have any considerable storage capacity. Regardless of the type of heater, the method of heat transmission is the same; i. e., the water is heated by coming in contact with the steam through the tubes. The total heat given off by a heater then is equal to the product of the area of the tube surface by the mean temperature difference between the steam and the water, and by a coefficient of heat transfer. The first two quantities can be determined for any particular case. The coefficient of heat transfer is a variable quantity and for these conditions varies from 100 to 350, in some cases under special conditions running as high as 1000 or more. The constant is dependent on four factors: 1. Velocity of the water through the heater. 2. Mean difference between the steam and water temperatures. 3. Material of the tube and its cleanliness. 4. Absence of air in the heater. Just to what extent each of these factors affects the final result is not definitely known. The velocity of the water in the tubes, for instance, is claimed by some to affect the result in direct proportion to its square root; other experimenters have published results where the constant varies as the cube root of the velocity.

A water-tube heater with copper tubes would appear to fill the requirements more perfectly than any other type of heater as the velocity of the water in the tubes is high and the heat-transmission rate of copper tubes is about 25 to 40 per cent greater than iron. The water-tube heater is most always an instantaneous heater and for that reason is not always applicable for every installation. Also a steam-tube heater is usually less expensive than a water-tube heater with a storage tank. The thermal efficiency of the heater is not impaired by a slower heat transmission rate. With the exception of the radiation losses on the outside of the tank, the water heater is nearly 100 per cent efficient.

There are a large number of different modifications of each type of heater on the market today. Some of them might be mentioned and described in detail, but the principle of operation of each type can be classified in one or the other classifications. The variations usually are brought out in the arrangement and size of coils, such as the straight, corrugated, specially-corrugated or U-tube heaters. Then there are the large coil and small coil heaters, each having its particular field of application. It is not the intention of this paper to recommend any particular type of heater, but to present an unbiased discussion and description of the problem encountered in the selection of this apparatus.

A type of heater which is used occasionally is one in which the steam and water come into actual contact. The Powers steam and water mixer is an example of this type of heater. A temperature controller is embodied in the mixer, and the capacity of the heaters is rated at 25 to 40 gal. per min. They are often used in industrial plants for supplying warm water in the wash rooms.

In selecting the size of the heater, it is best to refer to the tables of the manufacturers. These tables are usually accurate and can be depended upon. After the required storage and heating capacity of a heater has been determined, it will only be necessary to refer to the catalogue and to select a tank heater of the proper size and capacity.

The size of the steam-supply pipe necessary for the heater under consideration can be determined in a very simple manner. After the heating capacity of the coil is known, the total amount of heat necessary to heat this amount of water through a given temperature range can be found by multiplying the total water to be heated per hr. (expressed in lb.) by the mean temperature difference. Dividing the total heat necessary per hr. by the latent heat of the steam at the required pressure will give the total lb. of steam flowing per hr. Then the size of steam line can be selected from Table 3 showing the lb. of steam flowing in a given pipe at a given pressure with a 1 lb. drop in pressure.

TYPES OF HOT WATER REGULATORS

It is always advisable to install a hot-water regulator on every tank as the regulator will maintain a constant pre-determined temperature in the tank if properly installed. There are two different types of regulators on the market, in one of which the operating medium is a volatile liquid as in the Ideal Sylphon or the Powers; in the other, the operating medium is water or compressed air, as in the Weld. In the first type, the valve usually has graduated action; *i. e.*, the valve stem will stay in any position between full open or closed as long as the demand is such as to require just that amount of steam. The second type usually is operated by water and the valve action is positive, *i. e.*, the valve either is full open or closed. The graduated-action valve is probably best where the load is more nearly constant, as in this case the wire-drawing action on the valve seat is not so marked. However, both types have been used and have given excellent results in practically all types of installations. It eventually is a matter of personal opinion and experience.

One definite rule in selecting the size of regulators can be given, however, and that is to rather make the regulator one size too *small* than too large. A regulator which is too large will invariably cause wire drawing on the seat and eventually make any regulation impossible, whereas a regulator which is too small will only cause an additional slight pressure drop through the regulator, which is of no great consequence.

Another type of heater which is used a great deal in apartment houses and homes, is the gas heater. These heaters are usually of small capacities ranging in size from the small house heater to the larger heater, heating from 8 to 10 gal. per minute with a 60 deg. temperature rise. They are usually equipped with an automatic regulator and valve which opens the gas valve as soon as the water begins to flow and holds it open until the water is shut off. In addition to this valve, there is a thermostatic valve which cuts off the gas supply as soon as the water reaches a maximum temperature.

Simple gas heaters are limited to installations serving not more than four bathrooms and kitchen and laundry fixtures.

Electric heaters are a recent development and are only furnished in small units. There are two types, one called the immersion type, where the heating element is encased in a tube and inserted in the water pipe, and the other where the element is applied on the outside of an ordinary range boiler and covered with insulation.

The next item which comes under consideration is the design of the hot-water-supply piping. Correct design embodies five things: *A*. Correctly sizing the pipes to supply hot-water fixtures. *B*. Providing proper return pipes to produce circulation. *C*. Arranging both supply and return piping so as to prevent accumulation of air or to relieve the air. *D*. To make the air relief automatic. *E*. Providing for the elimination of corrosive action of the water.

ONE METHOD OF SIZING WATER-PIPING

There are various methods of sizing water-piping which are in use today. One of the best methods which has been devised was published in the *Metal Worker, Plumber and Steam Fitter,* on March 23, 1917, by H. L. Alt (see Table 4). His method is briefly this. He classifies all fixtures in four groups according to the supply-pipe sizes. Each group or class is designated with a conversion factor by which all fixtures can be expressed in the same terms. The groups are:

A. Fixtures with ⅜ or ½ in. supplies such as lavatories, flush tank closets, flush tank urinals and drinking fountains.

B. Fixtures with ⅝ or ¾ in. supplies such as sinks, slop sinks, showers, baths, etc.

C. Fixtures with ¾ or 1 in. supplies such as urinals with flush valves.

D. Fixtures with 1¼ or 1½ in. supplies such as flush-valve water closets, flush-valve slop sinks, etc.

Considering the Class *A* fixtures as unity, all other classes can be reduced to equivalent Class *A* fixtures by multiplying by 4, 9 and 24 for Classes *B, C* and *D,* respectively, so that any installation can easily be reduced to an equivalent number of Class *A* fixtures.

The above list of course includes fixtures that use cold water only, such as drinking fountains, closets and urinals which do not come under this discussion, but Table 4 is given complete as Mr. Alt published it, with the exception that the capacities of the various pipes are based on the relative delivery capacity and not on the relative area of the pipes.

TABLE 3. FLOW OF STEAM THROUGH PIPES—THE SIMS CO.

Initial Pressure by Gage lb. per sq. in.	Diameter of Pipe in In.—Length of Each 240 Diameters; Weight of Steam per Min. in Lb. with 1 lb. Drop: ¾	1	1½	2	2½	3	4
1	1.16	2.07	5.7	10.27	15.45	25.38	46.9
10	1.44	2.57	7.1	12.72	19.15	31.45	58.1
20	1.70	3.02	8.3	14.94	22.49	36.94	68.2
30	1.90	3.40	9.4	16.84	25.35	41.63	76.8
40	2.10	3.74	10.3	18.51	27.87	45.77	84.5
50	2.27	4.04	11.2	20.01	30.13	49.48	91.4
60	2.43	4.32	11.9	21.38	32.19	52.87	97.6
70	2.57	4.58	12.6	22.65	34.10	56.00	103.4
80	2.71	4.82	13.3	23.82	35.87	58.91	108.7
90	2.83	5.04	13.9	24.92	37.52	61.62	113.7
100	2.95	5.25	14.5	25.96	39.07	64.18	118.5
120	3.16	5.63	15.5	27.85	41.93	68.87	127.1
150	3.45	6.14	17.0	30.37	45.72	75.09	138.6

Initial Pressure by Gage lb. per sq. in.	Diameter of Pipe in In.—Length of Each 240 Diameters; Weight of Steam per Min. in Lb. with 1 lb. Drop: 5	6	8	10	12	18
1	77.3	115.9	211.4	341.1	502.4	1177
10	95.8	143.6	262.0	422.7	622	1458
20	112.6	168.7	307.8	496.5	731	1713
30	126.9	190.1	346.8	559.5	824	1930
40	139.5	209.0	381.3	615.3	906	2122
50	150.8	226.0	412.2	665.0	980	2294
60	161.1	241.5	440.5	710.6	1047	2451
70	170.7	255.8	466.5	752.7	1109	2596
80	179.5	269.0	490.7	791.7	1166	2731
90	187.8	281.4	513.3	828.1	1220	2856
100	195.6	293.1	534.6	862.6	1270	2975
120	209.9	314.5	573.7	925.6	1363	3193
150	228.9	343.0	625.5	1009.2	1486	3481

TABLE 4. SIZING OF WATER PIPES—H. L. ALT

Table of Pipe Sizes: Nominal Pipe Size	No. of Class *A* Fixtures Supplied	Classification of Fixtures H. L. Alt
⅜″	5	Class *A*—Fixtures with ⅜″ or ½″ supplies, such as
½″	10	lavatories, flush tank closets, flush tank urinals, and
¾″	20	drinking fountains.
1 ″	36	Class *B*—Fixtures with ⅝″ or ¾″ supplies, such as
1¼″	72	sinks, showers, baths, etc.
1½″	127	Class *C*—Fixtures with ¾″ or 1″ supplies, such as
2 ″	197	urinals with flush valve.
2½″	306	Class *D*—Fixtures with 1¼″ or 1½″ supplies, such as
3 ″	536	flush valve closets, flush valve slop sinks, etc.
3½″	758	One Class *B* fixture equivalent to 4 Class *A* fixture.
4 ″	1042	One Class *C* fixture equivalent to 9 Class *A* fixture.
5 ″	1833	One Class *D* fixture equivalent to 24 Class *A* fixture.
6 ″	2900	Table based on 50 to 100 lb. pressure; for pressures
8 ″	5380	below 50 lb. increase pipe one size.
10 ″	10000	Minimum size of mains and risers, ¾″.

THE DEMAND FACTOR

In sizing the piping, the demand factor must again be considered. It would be folly to size the piping for all the fixtures which are connected to the system, so a demand or utilization factor can be assumed for each class of fixtures. For instance, it would hardly be probable in a hotel building that all bath tubs would draw hot water at one time. It would depend on the size of the hotel, but the demand would hardly run over 50 per cent—more likely 33 1/3 per cent. In a school, however, it would easily be possible that all showers would be used at one time. Therefore this demand factor must be determined for each installation. The percentages given in Table 2 can be applied and good results can be obtained. After all fixtures have been tested as equivalent Class *A* fixtures, the piping is sized from Table 4.

WATER CIRCULATION

Circulating the water in the system is almost an absolute necessity, although circulation is not economical as far as fuel consumption is concerned, as about 10 to 25 per cent of the total fuel used in any system is used for circulation. If a circulating system is not installed, the water wasted by drawing the cold water out of the pipes and the poorer service which will result for the occupants of the building, make the omission of the circulating system inadvisable. The water may be circulated either by gravity or with a pump. In either case, it is advisable to design the system so that circulation will be natural. The best system and the most economical is the down feed system (see Fig. 1). In this case only one set of risers need be installed. The main supply riser is carried to the top story or attic and the risers are carried down through the building and the lower ends of the risers are connected to a circulating main and brought back to the tank. Sometimes, an installation is met with where an attic main cannot be installed, when an upfeed system must be installed as shown in Fig. 1. In this case, two sets of risers must be installed. The size of circulating mains is usually made from one-third to one-half the diameter of the supply main. Here again it is best to limit the size of risers and mains to ¾ in. as a minimum.

FIGURING PUMP SIZE

If the system of piping is complicated, covering a large area or with pockets in it, a circulating pump must be provided for. The pump must be of such size and capacity that it will circulate enough water to keep the temperature at the farthest outlet not lower than 10 deg. below the temperature at the tank. The total heat loss per hour in the entire system is computed and dividing this by the 10 deg. drop, the pounds of water to be circulated can be readily obtained. In order to obtain the head pumped against, the pressure of the cold water on the system must be known as the circulating pump must have at least 5 lb. excess pressure in order to circulate. In addition, the pump must circulate the right amount of water against the friction in the pipes. The sum of these two, i. e., the friction head plus the 5 lb. excess pressure will give the net head pumped against.

The third and fourth requirements of correct design, arranging both supply and return piping so as to prevent accumulation of air or so as to relieve the air and to make this air relief automatic, can be considered together. It is a well-known fact that the air will collect at the top of the system. The piping should be designed to pitch to a high point in the system where the air can be relieved. This is probably best ac-

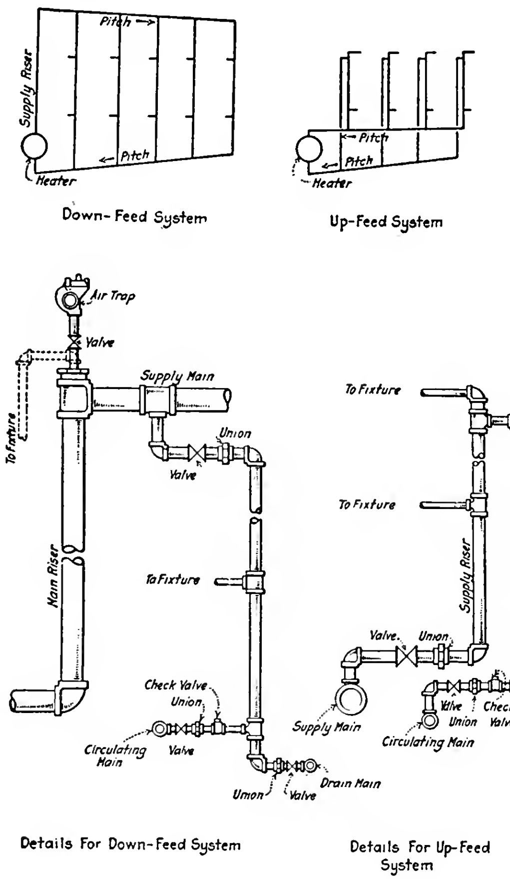


FIG. 1. DETAILS OF PIPING CONNECTIONS FOR BOTH DOWN-FEED AND UP-FEED SYSTEMS

complished by connecting a fixture supply at the top of the system so that the air can be drawn off through a fixture.

In case this cannot be done, an automatic air trap, such as an Anderson trap or a Kieley air trap, should be installed at the top of the system which will relieve the air automatically.

CORROSION OF PIPES

A phase of hot-water-supply piping which has claimed a great deal of attention in recent years is the incrustation and corrosion of piping. These are two separate and distinct chemical actions in the piping and each must be considered separately. Both subjects are large and I will not attempt to go into details. Briefly, incrustation is a chemical action which takes place in the water when it is heated. The lime and magnesia which are in suspension in the water when cold are precipitated when heated and scale up the pipes. The solution for this difficulty is the installation of a water softener. Frequently, water softeners are installed so as to soften the hot water of the system only.

Corrosion of the pipes is, as is now assumed by many, due to chemical action between the free oxygen in the water and the iron. It is this corrosive action which causes the pipes to "eat out." In recent years the only remedy which was considered practical was to install the piping of such material which would effectively resist this corrosion. Brass pipes, lead pipes or galvanized iron and steel pipes have been used in place of black pipes. Brass pipes are prohibitive at the present prices; lead pipe is not satisfactory because the hot water causes lead to expand but it will not contract when cooled, and this causes the pipes to sag. Galvanized iron and steel piping have been used mostly in recent years.

DEACTIVATING APPARATUS

There has been a great deal of information published on the relative corrosion of iron and steel pipe, but it is not within the scope of this paper to discuss this phase of piping. Experiments have been made by F. N. Speller, of the National Tube Co., to design a deactivating apparatus which would reduce the free oxygen content sufficiently to eliminate its corrosive action. The first deactivating apparatus brought out by Mr. Speller a few years ago was in the nature of an air-separating tank in which the oxygen was separated from the water. Apparently, this method did not prove sufficient as he has recently published another method in which the water is deactivated by passing through a tank filled with iron or steel slats on which the water may act and then by passing it through a set of filters, it is cleared. This subject has been discussed in recent issues of the JOURNAL of the Society. The subject of corrosion is one of considerable importance and where the corrosive action is great and no provision has been made to deactivate the water, the piping should be installed so as to be readily accessible for replacement.

EXAMPLE

In order to illustrate the method described above, a problem is herewith presented.

The building which has been assumed for this problem is a hospital and the number of fixtures are as follows:

Types of Fixtures	No.	Gal. per hour	Total
Private Lavatories	50	3	150
Public Lavatories	72	8	576
Slop Sinks	24	20	480
Baths	54	15	810
Showers	11	100	1100
Sinks	34	20	680
Laundry Trays	3	35	105
Laundry Washers	3	100	300
Maximum Connected Load			4201

The maximum water likely to be drawn at one time according to Table 2 would be 75 per cent of the maximum connected load or 4201 × 0.75 = 3150 gal. per hour. A storage capacity of 45 per cent of the heating capacity would be required or 1420 gal. Two hot-water tanks 48 in. × 96 in. were selected, each having a storage capacity of 754 gal.; with a low pressure coil in one of the tanks having a heating capacity of 2000 gal. per hour and a high pressure coil in the other tank having a capacity of 1160 gal. per hr.

The steam supply for the low pressure coil would be

$$\frac{2000 \text{ gal.} \times 8.33 \times (180 \text{ deg.} - 50 \text{ deg.})}{1043 \text{ B.t.u.} \times 60} = 34.5 \text{ lb. steam per sec.}$$

From Table 3, this would require a 4 in. steam supply. The size of the high pressure coil can be computed in a similar manner and would be 2 in.

By applying the conversion factors from Table 4, the total connected load would be equivalent to 653 class A fixtures and with a utilization factor of 75 per cent as assumed above, 653 × 75 per cent or 490 equivalent Class *A* fixtures are likely to be used at one time. From Table 4, the hot-water main would be 3 in.

Assuming a standard air cell insulation for the covering of the pipes with 100 deg. temperature difference between surrounding air and the hot water, a heat loss of 100,000 B.t.u. per hr. would take place in this problem. With a temperature drop of 10 deg. between the flow and return water 10,000 lb. of water would be circulated per hr. or 20 gal. per min. The friction head in this case was computed to be 20 lb. and this added to the 5 lb. excess over the initial water pressure would make a total pressure of 25 lb. or 60 ft. head. The circulating pump required would be a 1½ in. single stage centrifugal pump having a capacity of 20 gal. per min. against a 60 ft. head directly connected to a 2 hp. motor.

DISCUSSION

PERRY WEST (written): We shall all agree with Mr. Buenger, in what he has to say about the general dearth of published data on the subject. His comprehensive treatment contained in this presentation, should constitute the basis and stimulus for the collection of such data as would be most useful and gratifying to heating and ventilating engineers. I know personally of several engineers who have quite a collection of such experimental data, which will no doubt be made public in the near future.

In studying this paper I see that most of the vital points are touched upon, including:

1. The determination of demand rates and their bearing upon the design of the system.

2. The selection of hot-water generators best suited to the requirements and conditions.
3. The design and sizing of hot water supply piping and circulating system.
4. Incrustation and corrosion.

Properly exhausted, these four topics would constitute quite a complete treatise on the subject of hot-water service for buildings.

I should like to discuss each separately and at length, perhaps adding something here and there from the numerous experiments which I have had to conduct along the same lines, but time and space will permit only a passing discussion of the first three, as I wish to discuss more fully the last topic of incrustation and corrosion, which the author has only touched upon very lightly.

In regard to the demand rate I should like to emphasize the author's reference to this as being the fundamental factor upon which most every other factor connected with a hot-water system depends.

This is not only true of the hot-water system itself, but extends to all of its adjuncts, such as pumps, filters, softeners, or other treatment apparatus.

This is one of the first problems, therefore, which should be attacked by those who are seriously interested in any of these questions connected with domestic hot-water systems, and one upon which I have had some two dozen tests conducted, in buildings of various kinds within the past year.

While the author states that the usual methods of computing the probable demand for hot water in buildings and institutions of various kinds is not based upon such accurate scientific information as most of our other heating and ventilating computations, I believe that this is mostly true as far as the published data on the subject is concerned, and that experimental data is now in hand and in the course of compilation, which, when published, will form quite a complete array of such scientific information.

It is also found, by a thorough study of comparable data for different installations, that a great many of the variations in demand rates formerly attributed to indeterminate personal elements are traceable to other factors, or to personal factors which can be more or less accurately accounted for.

Table No. 2, submitted by the author as a basis for determining the maximum demand, is, I believe, intended to cover the maximum hour demand and not the maximum instantaneous demand, as this instantaneous demand is liable to run from 50 to 100 per cent above the maximum hour demand.

A table practically identical with this has been in use for some time in the office with which I am connected, and has been found quite useful for rough approximations of the maximum hour demand. Comparison with experimental data shows that the figures in this table are considerably higher in a great many instances than the actual requirements demand, and I am offering herewith a revised form—Table No. 5—showing certain corrections which I have found necessary to bring same into conformity with actual practice. A comparison of this table with the author's presentation will show that the quantity of water required for showers should be very materially decreased; that the quantity required for dish washers should be increased; and that the percentage of the total water for all fixtures likely to be drawn at one time, should be materially changed.

As stated above, such a table is very useful for a rough determination of the probable maximum hour demand rate, but cannot be depended upon to give very accurate results for any particular building or institution, unless a great many other factors peculiar to the particular plant are taken into consideration.

TABLE 5. HOT-WATER CONSUMPTION FOR VARIOUS TYPES OF BUILDINGS

Figured at a Final Temperature of 150° Fahr.

Gallons of Water per Hour per Fixture.

	Apt. House	Club	Gym.	Hospital	Hotel	Indust. Plant	Laundry	Office Bldg.	Public Bath	Private Res.	School	Y.M.C.A.
Private Lavatory	3	3	3	3	3	3	3	3	3	3	3	3
Public Lavatory	5	8	10	8	10	15	10	8	15		18	10
Bath Tubs	15	15	30	15	15	30			45	15		30
Dish Washers	15	30		30	30	30				15	15	30
Foot Basins	3	3	12	3	3	12				3	3	12
Kitchen Sink	10	20		20	20	20				10	10	20
Laundry Stat. Tubs	25	35		35	35		42			25		35
Laundry Rev. Tubs	75	75		100	150		100 to 150		100	75		100
Pantry Sinks	10	20		20	20					10	20	20
Showers	50	200	200	50	50	200			200	50	200	200
Slop Sinks	20	20		20	30	20	10	15	15	15	20	20
Dish Washers	300 Gal. per hour at 180° for serving capacity of 500 people											
Total Water for All Fixtures Likely to Be Drawn at One Time – %	20	50	80	60	50	90	100	15	100	50	25	75
Storage Capacity in % of Maximum Heating Capacity	100	75	50	50	25	50	25	100	50	100	50	50

I should like to speak particularly of these peculiar factors in connection with the various classes of buildings with which I have come into immediate contact.

In apartment houses, particular attention must be paid to the question of whether the showers are duplicate fixtures located over tubs, or are separate fixtures. Where located over tubs they may in most cases be entirely disregarded. Attention must also be paid to the question of the class of apartment, that is, if the apartment house is of a very high exclusive type, the number of fixtures per person is comparatively large, and the amount of water used per fixture would be correspondingly less if it were not for the fact that, generally speaking, the higher the class of tenant the more use he makes of plumbing fixtures, so that we find that in actual practice, the higher the class of building, the higher the rate of demand per fixture.

I have accordingly found it advantageous to divide apartment houses into three classes, that is, *A, B* and *C,* corresponding respectively to high class, ordinary and tenement class apartments. The ratio of the demand for these three classes runs about in the proportion of 100, 80 and 60.

Another factor which is found to have quite a bearing on the quantity of water used is the temperature at which the water is delivered, as between a range of 180 deg. and 125 deg. there may be a difference of at least 50 per cent in the total quantity of water used per day.

In connection with clubs, hotels, private residences and Y. M. C. A. buildings, somewhat the same allowance should be made for the class of buildings and occupants, also for the temperature of the hot water supplied, as is made for apartment houses.

In connection with office buildings, a very wide variation is found between those equipped with self-closing faucets and those equipped with compression bibbs. In a building equipped throughout with self-closing faucets, the demand rate may be as low as 50 per cent of that for the same building with ordinary compression faucets.

In connection with all classes of buildings, the habits of the people in the particular locality, or even in particular buildings of the same locality have a considerable bearing upon the hot water rate, and also upon distribution of the demand rate throughout the day. In industrial localities, for instance, the maximum demand periods will be confined almost entirely to the hours from 6:30 to 8:30 in the morning; from 11 a. m. to 1 p. m.; and from 5 to 7 in the evening. The washday demands will follow the early morning demand and rarely ever be superimposed on same.

In higher-class neighborhoods, the early morning demand occurs about an hour later, and is sometimes combined with the wash day demand, while there is also sometimes additional demands from 3 to 5 o'clock in the afternoon corresponding to tea time, and usually a demand between 9 and 11 o'clock in the evening, corresponding to the bath demand.

In very high class, exclusive neighborhoods the early morning demand is still later so as to almost always combine with the wash day demand, and besides the other demands there is usually an after-theatre demand extending to about 1 a. m.

By thoroughly studying all of these conditions, most of the indeterminate factors referred to by the author may be largely eliminated.

It may be interesting to the members to know that considerable work is being done along the lines of working out a more rational formula than Table 2 (see p. 166) affords, somewhat as follows:

First: To determine the maximum hour demand with reference to the total number of ordinary hot-water fixtures, the total number of persons using the fixtures, the number of users multiplied by the number of fixtures, the number of apartments and the number of rooms for various classes of buildings, together with the maximum hour rates to be added for kitchens in clubs, hotels, Y. M. C. A. buildings, etc., for laundries in hotels, private residences, hospitals, etc.; the maximum hour rates for kitchens being based upon the number of sinks and dish washers, and the maximum hour late for laundries being based upon the number of revolving washing machines and stationary tubs, and the number of pieces laundered per day.

I am submitting herewith Table 6, showing the actual maximum hour demand rates and total daily consumption for a number of buildings of different classes with the various rates worked out for the factors mentioned above. It will be noted that there is quite a regularity in the amount of hot water used per day per fixture for buildings of the same type and class, and that there is also a very regular relation between the total quantity of hot water used per day and the percentage which the maximum hour demand bears to this total quantity. There is not, however, any very great degree of regularity for the other factors mentioned, so that we have come to regard these two factors as being the most reliable for use in estimating the total quantity of hot water used per day and the maximum hour demand rate.

It will be seen that the total consumption of hot water runs about 25 and 20 gal. per day per ordinary fixture for the Class *A* and *B* apartments respectively, and it is found to run about 15 gal. per day per ordinary fixture for the Class *C* apartments; also that the maximum hour demand equals about 10 per cent of this total in each case. It should be noted that all of the figures in Table 2 are on a basis of 100 per cent occupancy of the building during maximum winter loads. Care should be exercised, in running the tests and making comparisons, to ascertain these conditions.

For office buildings the total consumption runs about 50 gal. per fixture per day for ordinary buildings with public toilets and practically no plumbing in the individual offices; about 15 gal. per fixture per day for buildings with individual fixtures in the offices; about 50 per cent of these amounts may be taken for buildings with self-closing faucets. In most all cases the maximum hour demand rate will approximate very closely 10 per cent of the total daily consumption.

For hotels the total consumption for the house outside of kitchens and laundries, runs about 80 gal. per ordinary fixture per day for large transient plants, and the maximum hour demand equals about 7½ per cent of the total. For apartment hotels, the total consumption for the house runs about 50 gal. per fixture per day, and the maximum hour about 5 per cent of this total.

Hotel kitchens take about 2500 gal. per day for a continuous operation tank type dish washer operating on an average of 8 hr. per day, and the maximum hour demand for a kitchen equals about 20 per cent of the daily total including dish washers, sinks, etc.

Laundries consume about 1 gal. of hot water per piece of laundry handled, and the demand rate is fairly even during the hours operated. The average size washing machine consumes on an average of about 150 gal. of hot water per hr.

In regard to the selection of apparatus it will be noted that Table 2 calls for storage tanks from 35 to 50 per cent of the maximum hour demand. This means that the capacity of the heater would require to be equal to the maximum hour demand, otherwise the storage of hot water

TABLE 6. DATA ON HOT-WATER CONSUMPTION IN BUILDINGS FOR 100 PER CENT OCCUPANCY

Class	Buildings	Number of Fixtures: Baths	Basins	Kitchen Sink	Slop Sink	Laundry Tub	Shower	Total Without Shower	Total With Shower	Number of: Apts.	Rooms	Occup'ts.	Total Hot Water	Gallons H.W. per Day for Ordinary Fixtures Exclusive of Laundry Machines & Hotel Kitchens — Hot Water per: Fixture Without Shower	Fixture With Shower	Person	Person per Fixture Without Shower	Person per Fixture With Shower	Apt	Max Hr	Max Hr % of Day
A	12 Story Apartment West	271	500	220		140	166	1131	1297	66	1012	553	23000	20.3	17.8	41.6	.0368	.0323	383.0	2100	11.0
	12 Story Apartment East	159	288	88		77	126	612	738	37	637	353	10000	16.4	13.6	28.4	.0463	.0386	270.0	1000	10.0
	18 Story Apartment	131	230	97	32	99		589		33	508	227									
	12 Story Apartment	93	176	41	2	77	66	389	455	23	301	148	13000	33.4	28.6	88.0	.0249	.0193	565.0	1350	10.4
	14 Story Apartment	94	120	55		45	39	314	353	28	329	197									
	12 Story Apartment	132	132	96	48	30		438	438	49		200	10700	24.4	24.4	53.5	.0122	.0122	218.0	970	9.1
	13 Story Apartment	101	160	52		48	72	361	433	27	499	189	8600	23.8	20.0	45.5	.0126	.0106	319.0	840	9.8
	12 Story Apartment	102	137	75		94	45	408	453	61	521	268	9500	23.2	21.0	35.4	.087	.078	156.0	1200	12.6
	13 Story Apartment	153	163	147		178		641	641	94	473		14585	22.7	22.7				155.0	1163	8.0
	Average	137	212	97	27	88	86	453	601	46	535	267	12769	23.5	21.2	48.7	.0366	.0318	295.1	1232	10.1
B	12 Story Apartment	49	61	50		52	48	212	260	49	266	125	4800	22.6	18.5	38.4	0.170	0.148	99.0	450	9.4
	12 Story Apartment	37	62	38		65	36	202	238	37	244	135	3660	18.1	15.2	27.1	0.134	0.113	99.0	270	7.4
	8 Story Apartment	33	74	34		66	32	207	239	33	236	143	3000	14.5	12.6	21.0	0.101	0.088	91.0	390	13.0
	Average	40	66	41		61	39	207	246	40	249	134	3820	18.4	15.4	28.8	0.135	0.116	96.0	370	9.9
A - Transient	Central Plant for 500 Houses	500	500	500		1000		2550			4000	5000	53110	20.4		17.7	0.0069			5023	9.5
	13 Story Hotel	385	535	100	105	60	13	1183	1206		660	1382	97350	82.0	81.0	70.0	0.595	0.585		7400	7.6
	14 Story Hotel	540	733	10	55		75	1343	1418		600		115000	85.5	81.0					8625	7.5
	Average	462	635	55	80	30	44	1263	1312		630	1382	106175	85.7	81.0	70.0	0.595	0.585		8012	7.55
B	12 Story Apt. Hotel	155	170	7	10		154	342	496	170			16252	47.5	33.0				95.5	650	4.0
	22 Story Office and Terminal	6	2924		50	Soda Fixtures 10	10	2990	3000		160[illegible]	11000	44540	14.9	14.9	4.5	0.00135	0.00135		4420	10.0
	10 Story Office Bldg with Self-Closing Faucets			256			1	255	256		235		1545	6.5	6.5					154.5	10.0
	16 Story Office Bldg.		72	Misc 35	50		2	157	159			7000	7500	47.7	47.7	1.07	0.0068	0.0068		750	10.0
	8 Story Gov't. Warehouse		87		46			133	133				6000	45.0	45.0					361	6.0

would be exhausted within from 20 to 30 min. and the tank be filled with water below the normal temperature during the remainder of the demand period. This heavy demand period for apartment houses rarely exceeds an hour, except on wash days, when it may continue for from three to four hours. In office buildings the demand rate is regular and in hotels the rate is quite uniform over long periods.

For apartment houses the storage apparatus should be increased therefore, and may be decreased somewhat for office buildings and still more for hotels.

The average demand rates compared with the maximum hour demand rates for these three classes of buildings, are about 50 per cent for apartments, 60 per cent for office buildings and 70 per cent for hotels, so that it would seem that the best practice would be to have a storage capacity equal to the maximum hour demand, and a heater capacity equal to 50 per cent of this for apartments and office buildings, and a storage capacity of 25 per cent and a heater capacity of 75 per cent for hotels.

In regard to the selection of heaters, the author has pointed out that some cast-iron boilers are too lightly constructed to withstand high pressure, and it might be added that our existing boiler codes limit the working pressures for these. On the other hand it should be borne in mind that steel heaters are, on account of their thinner and more vulnerable material, much more subject to destruction by corrosion, unless proper steps be taken for their protection. A steel heater, properly protected, however, makes an ideal installation.

Copper tubes corrode much slower in contact with ordinary hot water than do brass tubes, and are therefore much to be preferred in heater construction.

The author has pointed out that an adequate circulation system is essential to the satisfactory and economical operation of any hot-water service system. It may be added that the better a system is in this respect and the faster the circulation, the more rapid the corrosion, other things being equal. The cure for this, in the light of our modern engineering science, is not to impair the usefulness or economy of the system for fear of installing an adequate circulation system, but to apply the proper methods for the prevention of this corrosion.

The subject of incrustation and corrosion have been well touched upon by the author but not treated as exhaustively as the other subjects.

These subjects are somewhat correlated since incrustation tends to retard corrosion, while whatever corrosion there is tends to add to the amount of incrustation. Generally speaking, a hard water is one containing appreciable quantities of the hydroxides, bicarbonates, sulphates, or chlorides of calcium, magnesium and barium in solution. Heating the water to the temperature of ordinary hot water service breaks up some of the bicarbonates and chlorides, but has little or no effect upon the sulphates. The soluble bicarbonates are broken up into insoluble monocarbonates, which are deposited upon the interior surfaces of the heaters and pipes. These deposits tend to scale up and reduce the effectiveness of heating surfaces and to choke the pipes, especially small branches. On the other hand these same deposits tend to form a protective coating

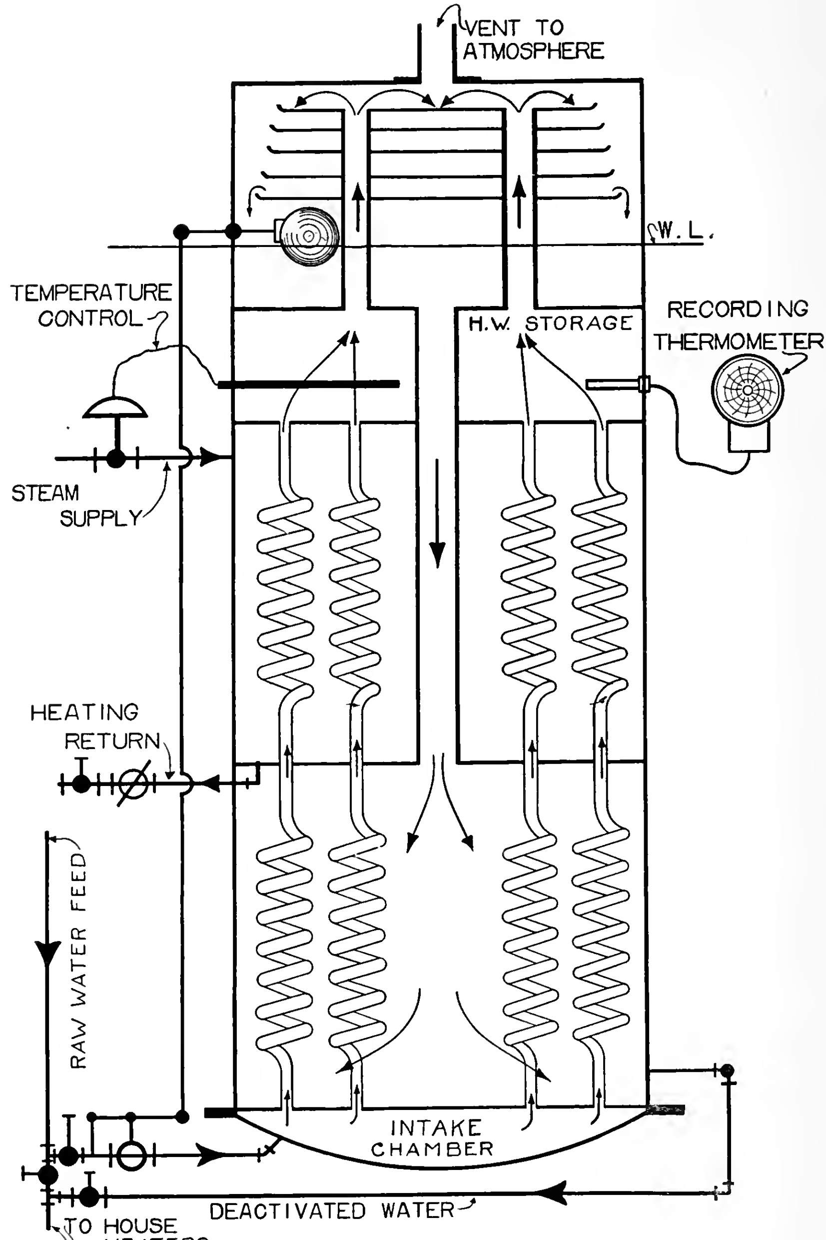


FIG. 2. DIAGRAMMATIC CROSS-SECTION OF DE-AERATING DEACTIVATOR

which retards corrosion and in some extreme cases practically prevents any appreciable deterioration from this source. The mere presence of some of these alkaline compounds in the water also tends to reduce corrosion due to their de-ionizing effect. For these reasons some of the strongly alkaline and limestone waters found in this country are practically non-corroding in their effect upon a hot-water service system.

Most waters met with in practice, however, are soft enough naturally, or are artificially treated so as to offer practically no resistance to the rapid corrosion which takes place in hot-water service systems.

The softening of hard waters by the removal of the scale forming elements is highly desirable from the standpoint of reducing the scaling of heating surfaces and the choking of pipes, etc.

In addition to the practice of softening water for the purpose of preventing scale as referred to by the author, it is frequently advantageous to soften water intended for use in laundries, hotels, industrial plants and even in apartment houses and dwellings, for the purpose of making it more pleasant to use and more economical in the quantity of soap required for washing, etc.

In the case of fairly soft water, or in the case even of a hard water, where most of the hardness is due to the sulphates or other non-alkaline and non-encrusting ingredients, this softening process has little or no bearing upon the corrosion which this water will cause. If, on the other hand, the water to be softened, contains an appreciable amount of hardness due to alkaline or encrusting ingredients, the removal of these in the softening process prevents the encrusting or alkaline protection referred to above, and therefore allows corrosion to proceed much more rapidly with the softer water than would be the case with the untreated hard water.

Some of the chlorides, especially magnesium chloride, will hydrolize and partially dissociate into magnesium hydrate and hydrochloric acid, which acid combines with the metal of the pipes, etc., to form a metallic chloride, setting free hydrogen. The amount of hydrochloride acid thus set free in ordinary cases is very small, and if allowed to act but once and then come to equilibrium, would cause comparatively little corrosive trouble.

If free oxygen be present in the water, however, which is usually the case unless particular pains have been taken to remove it, this free oxygen combines with the hydrogen set free, forming water and depolarizing the electrolytic circuit, thus allowing the process of corrosion to proceed. In the meantime the new metallic chloride formed re-hydrolizes, forming the metallic hydroxide and setting the hydrochloride acid free again. The process will then begin all over again and continue as an indefinite cyclic process, so long as there is enough free oxygen present to keep the electrolytic circuit depolarized. As new water is added additional hydrochloric acid is set free and if allowed to concentrate, as in boilers, is one of the most dangerous corrosive elements met with in practice.

In the case of sea water which is sometimes heated for bath purposes, this action is very marked and can only be retarded by neutralizing the acid or by the removal of the free oxygen from the water.

The sulphates are not materially affected by the temperatures of ordinary hot service water, and have no particular bearing upon incrustation or corrosion in these systems. It should be noted, therefore, in this connection that the hardness due to sulphates in a water does not neces-

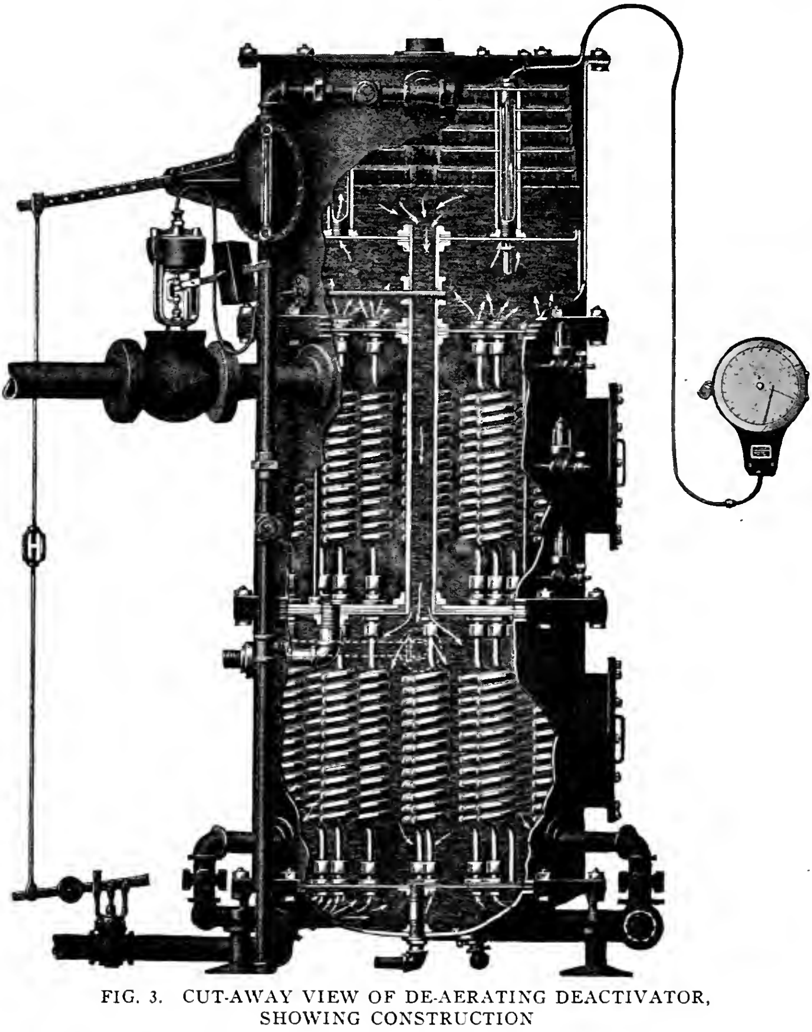


FIG. 3. CUT-AWAY VIEW OF DE-AERATING DEACTIVATOR, SHOWING CONSTRUCTION

sarily indicate that such a water will precipitate any scale forming protective coating in the pipes and boilers unless the carbonates be present also.

It will be seen that soft waters and such hard waters as contain small quantities of bicarbonates are not scale forming in ordinary hot water systems and do not therefore tend to protect against corrosion in this way.

The foregoing gives a fair idea of the relationship between the chemical properties of the water and the subjects of incrustation and corrosion,

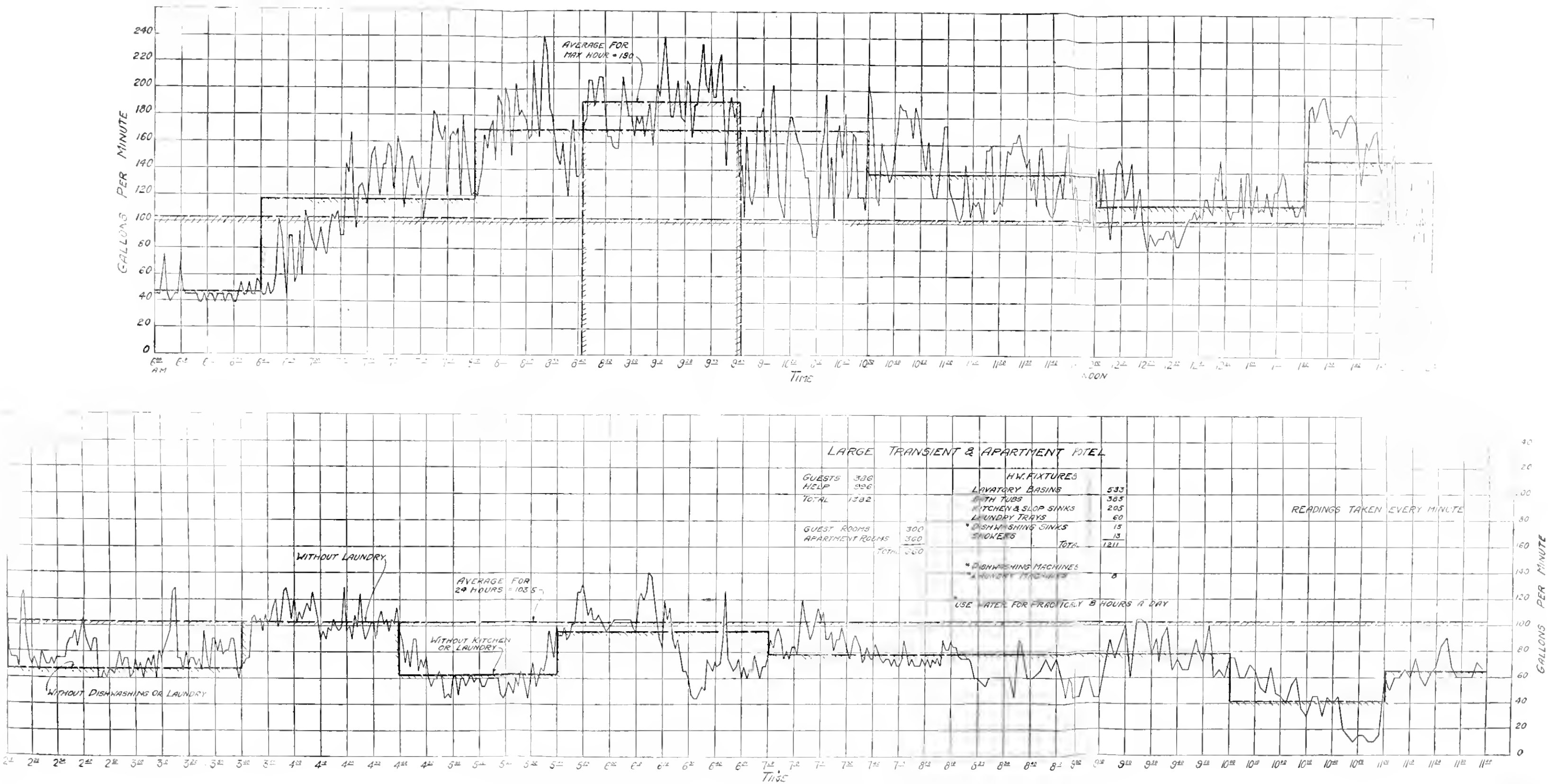


FIG. 4. CHART SHOWING INSTANTANEOUS HOT-WATER DEMAND RATE FOR HOTEL CONTAINING 660 ROOMS

from which it may be seen that a good chemical analysis of the water is a very helpful factor in designing a hot water system.

As the author has stated and as has previously been demonstrated on numerous occasions by the leading authorities, who are making a study of this subject, practically all of the corrosive activity between the water and the metals of a hot-water service system is due to the presence of free dissolved oxygen in the water. This action is sometimes intensified by the presence of chlorides as referred to above in the case of magnesium chloride, or by the presence of appreciable quantities of free, dissolved, carbonic acid gas in the water. The action of this free carbonic acid gas is mainly along the lines of giving the water an acid tendency which promotes the electro-chemical corrosive activity much in the same way as hydrochloric acid, except in a much milder degree.

In other words it is a well-established fact now that no appreciable corrosive activity will ensue between the water and the metals of an ordinary hot-water service system unless the water contains an appreciable amount of free dissolved oxygen or is contaminated with acids or other corroding elements foreign to ordinary potable waters. Metals will not corrode in contact with H_2O without the additional presence of free dissolved oxygen.

It is also a well-established fact that the removal of this free dissolved oxygen down to a reasonable degree will prevent this source of corrosion.

The theory of this corrosive activity is now so well established and perhaps so thoroughly understood by most of the members present, and is also so fully vouched for by the leading chemical authorities who have made a study of this subject, such as Cushman, Walker, Speller, etc., that it is not thought advisable to enter into any lengthy argument or discussion of this phase of the subject.

It may be of interest to recount some of the technical points connected with this ordinary cause of corrosion.

This corrosive activity being an electro-chemical action involving hydrolizing and ionizing, naturally takes place much faster at high than at low temperatures. The corrosive activity in hot-water systems is, therefore, from five to ten times as rapid as in cold water systems, other conditions being equal. In the case of sea water the magnesium and sodium chlorides augment the action so as to make corrosion with cold salt water almost as serious as with hot fresh water.

Ordinary potable waters, with the exception of some deep well, or other ground waters, contain on an average about 5 per cent of dissolved air. Of this there is usually about 20 per cent oxygen, which means about 1 per cent of oxygen by volume at ordinary temperatures and atmospheric pressure. Much higher saturations may be reached under higher pressures such as are generally found with air pressure tank systems. Every lb. of this oxygen may be responsible for at least 3 lb. of corrosion, which means that for every 1000 gal. of water about 0.25 lb. of iron or steel may be eaten away by this corrosion. Inasmuch as the useful life of ordinary iron or steel pipe is practically ended by pitting when 20 per cent of its weight is corroded away, the rate of destruction in a hot-water service system may run as high as 1.25 lb. per 1000 gal.

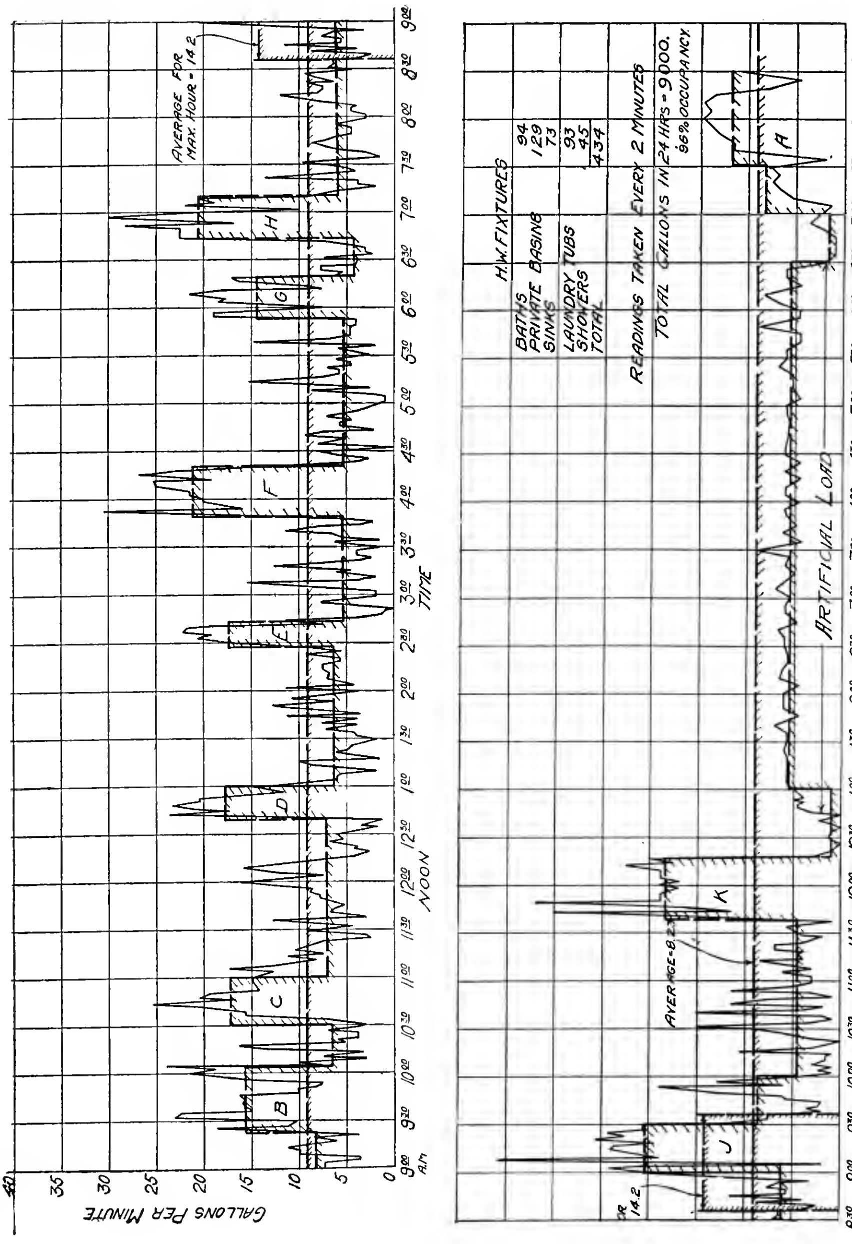


FIG. 5. CHART SHOWING INSTANTANEOUS HOT-WATER DEMAND RATE FOR 12-STORY HIGH-CLASS APARTMENT BUILDING

of water. This would mean the loss of about 1000 lb. per year for an ordinary size building, which would represent a replacement cost of about $2000. It is estimated that the annual loss occurring in this way in hot-water service systems throughout the United States amounts to $50,000,000.

As the author has suggested, the way to prevent this corrosion is to eliminate this free oxygen, and you will no doubt be interested in some of the latest progress in the development of commercial apparatus for this purpose.

Basically there are two methods which may be designated as the deoxidizing method and the deaerating method. The deoxidizing process is carried on by allowing the hot water to pass from the heater into a tank or deactivator where it is brought into contact with thin sheets of perforated metal especially prepared and arranged to allow the water to exert its corrosive activity upon this comparatively cheap and easily replaceable metal instead of upon the more expensive and less easily replaceable materials of the piping system. The water is then passed through a hot water filter to free it from the objectionable discoloration caused by the products of corrosion in the deactivator, and is ready for use in the system without danger of any further corrosion or discoloration.

The deaerating process is carried on by removing the free dissolved gases from the water by heating, by vacuum, or by a combination of the two while the water is in a highly divided state brought about by spraying or cataracting over baffles in a deactivator. This process, as will be easily understood, removes not only the oxygen, but the other free dissolved gases, such as CO_2, nitrogen, etc.

The removal of all gases eliminates some of the air binding and splashing difficulties referred to, besides stopping corrosion.

Water thus treated by either process for the removal of its active corroding elements is sometimes termed deactivated water, meaning that it is minus its corroding activity.

The removal of oxygen by the deoxidizing process and the removal of the gases by the deaerating process amounts to from 90 to 99 per cent of the quantities contained in the raw water in some of the commercial apparatus now on the market.

Numerous experiments have shown that the rate of corrosion is practically in proportion to the percentage of free oxygen present, so that a removal of 90 per cent means an increase of tenfold in the life of pipes and other apparatus protected.

If the average useful life of iron or steel hot-water service systems using raw water be taken at from 7 to 10 years, the average life of the same system, using 90 per cent oxygen free water, would be from 70 to 100 years. Inasmuch as this is far beyond the life of the average building a removal to this point is considered quite sufficient for this kind of service.

If the average useful life of a brass system be taken at from 20 to 30 years with raw water, the useful life of the same system using deactivated water might be from 200 to 300 years.

The rusting of pipes not only causes deterioration, but on account of the rust occupying from 10 to 20 times the space occupied by the metal from which it originates, causes a very rapid choking and stopping up of the pipes. Rust also discolors the water, giving it a very disagreeable appearance, and in turn causes very unsightly stains which are hard to remove.

You will also be interested in knowing something of the cost of such apparatus as we have been talking about, and as to what return it may be expected to show upon the investment.

Ordinarily speaking, the cost of this apparatus will run from 5 to 10 per cent of the replacement cost of the materials protected. This means that the carrying and depreciation charges might amount to from ½ to 1 per cent of the replacement cost. Figuring on an average life of ten years for piping and apparatus without protection, it would mean that for a total cost of from 5 to 10 per cent of the cost of replacement the entire system may be protected and such replacement costs postponed practically indefinitely.

Another way of measuring the cost is on the basis of amortizing the cost of the apparatus over a period of 10 years. The sinking fund cost together with the carrying charges would amount to from ¾ to 1½ per cent per year, and at the end of 10 years the entire first cost and carrying charges would be wiped out, thus giving a full paid up insurance for the remainder of the life of the building.

This whole proposition of arresting and preventing corrosion in the piping systems of buildings may be considered as another important item which has been added to the general conservation and insurance movements.

It has been the custom for years to insure buildings against loss by fire and other destructive agencies, and we have, of course, had our forestry system for the preservation of forests, and such other great movements as the conservation of coal, the enormous hydro-electric developments and the super-power plant scheme for the eastern seaboard, but still the pipes in our buildings, which are as important as the arteries in man, or the teeth in a horse, have been going without protection.

The protection of these pipes by the use of deactivated water is more complete, if you please, than any of the above mentioned protective and conservation undertakings for the simple reason that it protects absolutely, and almost indefinitely, while insurance on buildings is only a partial protection, and our usual conservation methods will not protect or reduce consumption beyond a certain relatively small percentage.

You will no doubt be interested in some illustrations of some of the commercial deactivating apparatus which is now on the market and also some of the curves showing characteristic hot water demand rates of the various types of buildings.

Figs. 2 and 3 illustrate a new type of deaerating deactivator, the construction and operating principles of which are as follows:

The various parts illustrated by these two figures are lettered to correspond:

A is an exchanger section, *B* is a heater section, *C* is a deaerating section, all being cylindrical in form and superimposed one above the other.

These sections are all constructed of 5/16 in. open hearth steel and

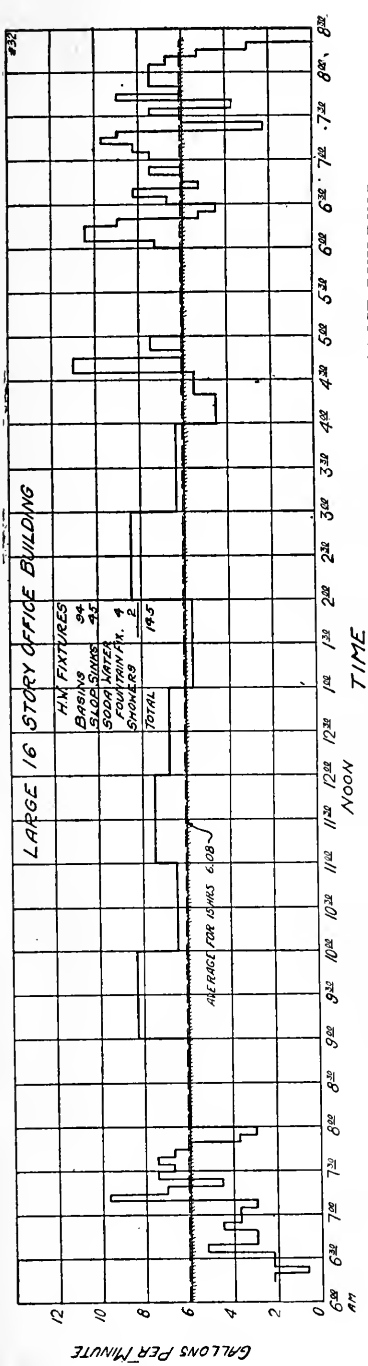


FIG. 6. CHART SHOWING THE HOT-WATER DEMAND FOR A LARGE 16-STORY O FFICE BUILDING

sections *A* and *C* are lined on the inside with regular chemical apparatus high temperature enamel for protecting all of the steel interior of these shells, which comes into contact with hot water.

The remainder of the apparatus is constructed either of copper or bronze, so that the entire machine is practically non-corrosive throughout and has a life far in excess of that of the ordinary buildings.

The raw water to be deactivated is brought in through pipes *D* and *E* to a distributing chamber *F*, from which it passes through the copper exchanger coils *G* into copper heater coils *H*. From these coils the water passes into hot water storage chamber *I*, from which it is led through pipes *J* to the top of a series of copper pans *K*, over which it is cataracted and falls into chamber *L*. From chamber *L* the deactivated water passes down through copper pipe *M* into the shell of exchanger *A*, where it passes over the outside of coils *G*, and out through outlets *N* and pipe *P* to the house heaters.

The operation of the apparatus is as follows:

The entering water is heated in coils *G* by an exchange of heat between the incoming and outgoing water so as to bring the temperature of the incoming water up to a certain point without the use of any heat except that given up by the outgoing water, thus reducing the temperature of this outgoing water. The water then passes into the heating coils *H*, where it is heated by steam in the shelf of section *B*, this steam being admitted through pipe *R* and controlled automatically by diaphragm valve *S*, which is operated by temperature element *T* in hot water storage chamber *I*, the condensation return being removed through pipe *U*.

The water thus being heated to the proper temperature for deactivation, is allowed to cataract over the pans where it is broken up so that the free dissolved

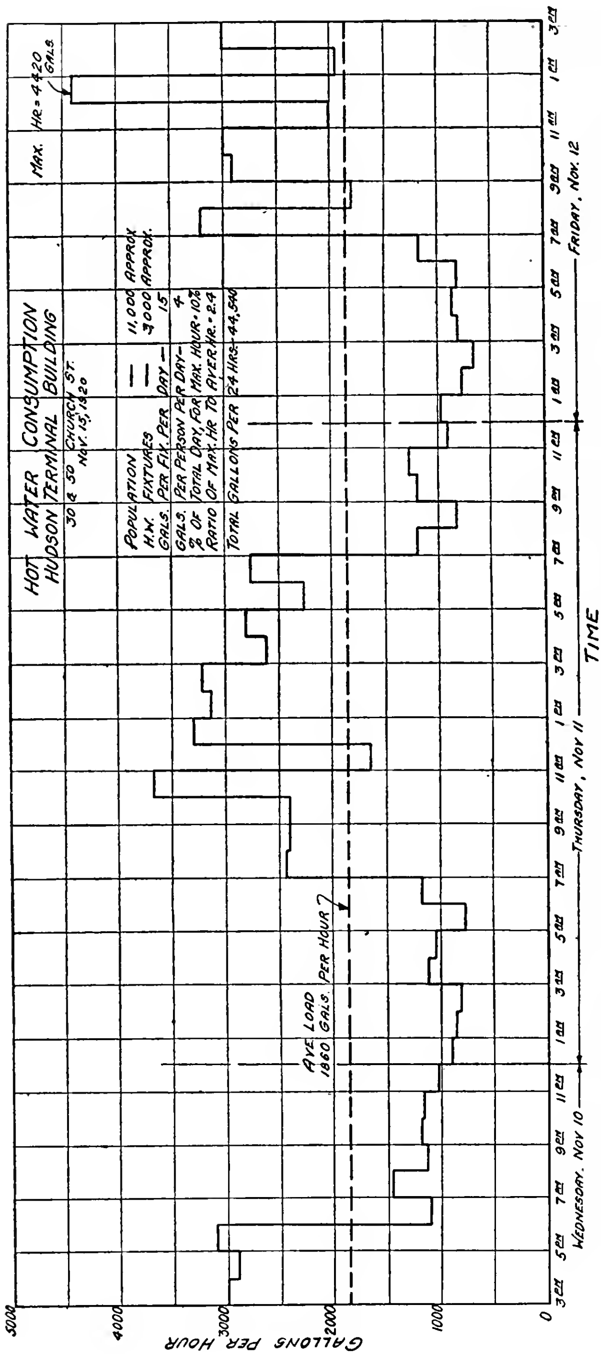


FIG. 7. CHART SHOWING HOT-WATER DEMAND FOR LARGE 23-STORY TERMINAL AND OFFICE BUILDING

gases are given up and are drawn out through copper vent *V*, which is extended up to the atmosphere and capped with a ventilator to give the proper flue action for the removal of these gases.

The removal of these free dissolved gases by this type of apparatus is due to the fact that the saturation contents of water is automatically reduced as the temperature is raised, so that at a point approximating the boiling temperature about 95 per cent of these gases will be liberated. The deactivated water then passes out through the exchanger section as explained and goes to the house heaters about 50 deg. above the temperature of the incoming water. This 50 deg. raise represents the net heating which is required to be done in this apparatus, although the temperature at which the water is raised in the deaerating chamber is about 210 deg.

All of this heat which is added to the water leaving the apparatus is, of course, transferred directly to the system and reduces the amount of heat necessary to be added by the house heaters, thereby causing no loss of heat except a very small percentage for radiation and allowing the machine to operate at a thermal efficiency of almost 100 per cent.

The water to be deactivated is fed to the machine through a balanced valve *W*, which is operated by a float *X* in the deaerating chamber, so as to maintain a constant water line in this chamber by feeding water to the apparatus at a rate commensurate with the rate at which water is drawn from same to the house. The recording thermometer *Y* is connected to the hot water storage chamber *I* so as to make a record of the temperature at which the water is being deactivated.

Shut-off valves are usually installed on the inlet and outlet of the machine and a valved bypass is furnished so that the machine can be shut off for repairs and cleaning without interrupting the house service. Valved blow-downs are furnished for the various sections so that the apparatus may be easily cleaned and drained. Large manholes are furnished for inspection and repairs.

Fig. 4 shows the instantaneous hot-water demand rate for a large 15-story high class transient and apartment hotel, containing 660 rooms. Readings were taken every minute over a period from 6 a. m. to 11:40 p. m. The average load on this building for the entire time was 103.5 gal. per min., and the maximum hr. load occurring between 8:40 and 9:40 a. m. was 190 gal. per min. The total water consumption per day of 24 hr. amounted to about 150,000 gal. The total daily consumption for ordinary fixtures, exclusive of kitchens and laundry, was about 100,000 gal.—the laundry consisting of 8 machines operating 8 hr. per day consumed about 20,000 gal. and the kitchens having a total of about 15 dish washing sinks and one large spray type dish washing machine consumed about 30,000 gal. The minimum load occurring between 10 and 11 p. m. was 40 gal. per min.

Fig. 5 is for a 12-story high class apartment house containing 48 apartments, and showed an average daily consumption of 8.23 gal. per min. over a period of 24 hr., readings being taken every 2 min. during this time. The maximum hr. load occurred between 8:30 and 9:30 a. m. and amounted to 14.2 gal. per min. Some of the instantaneous loads, as will be noted, ran as high as 130 per cent of this maximum hr. reading. This curve gives some idea of the wide fluctuation and the various number of loads which occur in a house of this character, as follows:

A is the breakfast and dishwashing load,
B and *C* are the wash day load,
D is the luncheon load,
E is the lunch dishwashing load,
F is the afternoon tea load,
G is the dinner load,
H the dinner dishwashing load,
J the bath load,
K the after-theatre supper load.

Fig. 6 is for a large 16-story office building, having most of the hot-water fixtures in public toilets and very few in the individual offices.

Fig. 7 is for a large 23-story terminal and office building, having the greatest percentage of hot-water fixtures in the individual offices, in addition to the hot-water fixtures in the public toilets.

WM. WILCOX (written): After this subject has been so capably treated as in the paper just presented, I hesitate to make any comment, more particularly because the method of determining demands has been followed by such a large number of architects and engineers.

I have, however, the conviction that hot water needs should be determined upon the unit of service rather than upon the flow capacity of any particular fixture. An outlet may flow at the rate of 2 gal. or 20 gal. per min. but if it is only used for a limited number of sec. in an hr. or a day, there is not much profit in determining the amount which will flow per min. and basing calculations upon the information obtained.

I think that there is a general agreement that 1 gal. will provide sufficient hot water at 180 deg. at the heater to take care of one lavatory service. If as is frequently the case, the hot water is used by holding hands beneath the faucet, where a good circulating system is installed, the water will become too hot for use before a pint has been used. In different types of buildings similarly one can estimate very closely on the quantity of water which will be taken for a bath tub and probably less accurately for the amount used for each shower bath. With the known occupants of a building, the probable number of times each fixture will be used per hour can be very closely estimated.

Estimates made upon the basis of the method I will show, call for a quantity of water much smaller than estimates made upon the basis of fixture flow, but in all cases where a check has been made, an adequate quantity of water has apparently been provided. By the estimate sheets and charts, I hope to show as undesirable an arbitrary assumption of relative storage and heating capacity and believe that in many hotels, apartment houses and industrial establishments it will be found that storage capacity should be from 100 to 500 per cent of hourly heating, instead of 30 to 50 per cent of hourly heating. The estimate sheets referred to also provide for the particular requirement of each fixture for any one hour and the peak of one fixture may occur at a time different from the peak of the other fixtures. This could not be provided for in a fixture table as it can be in an estimate sheet covering each hour of the day.

300 ROOM HOTEL — HOT WATER ESTIMATE SHEET — Form No. 101 #1

These ratings are based on hot water at 180° F

	Showers Std 4" head 10 gal. per bath-6 min.	Bath tubs [illegible] gal. per bath 142	Kitchen taps 5 gal per hour 15	Slop sink taps 5 gal per hour 16	Hand lavatories 1 gal per service 184	DISH WASHER	COOKING				Pounds of Steam used per hour	Total required Gals per hour	Gallons heated per hour	Hourly surplus Gallons	Total surplus Gallons
1 A. M.	MAXIMUM HOURLY STEAM REQUIRED 960 LBS = 32 HP = 3840 □' RADIATION														
2	STORAGE CAPACITY 125% OF HOURLY HEATING														
3															
4															
5							20				24	20		+780	1000
6		160			20		60				288	240		+560	1000
7		600	50		50	20	40				885	760		+40	1000
8		1120	75	75	100	80	20				960	1470		-670	330
9		520	100	100		100					960	820		-20.	310
10				100							120	100	800 GALS PER HOUR	+700	1000
11			50	100	30		20				240	200		+600	1000
12 M.			80	100	100	20	60				432	360		+440	1000
1 P. M.			100		100	120	40				432	360		+440	1000
2			75			180	20				330	275		+525	1000
3						100					120	100		+700	1000
4		100									120	100		+700	1000
5		250			20		30				360	300		+500	1000
6		320	50		50	20	60				600	500		+300	1000
7			100		150	50	40				408	340		+460	1000
8			100		100	150	20				444	370		+430	1000
9		600				100					840	700		+100	1000
10		900									960	900		-100	900
11		640									748	640		+160	1000

12 Steam Pressure 5 LBS

Steam available from HEATING BOILER

Water Pressure 40 LBS.

No. 1 (Estimate Sheet) shows in different columns the estimated hourly requirements for showers, tub baths, kitchen taps, slop sinks, hand lavatory, dish washer and cooking, from which we see that the total storage plus hourly heating must be sufficient for the greatest peak load. In this particular estimate we have assumed an hourly heating capacity of 800 gal. which means a maximum steam consumption of 960 lb. equal to 32 hp., the equivalent of 3840 sq. ft. of radiation. During most hours of the day the demand does not exceed the hourly heating, therefore, the storage of hot water is not materially reduced. In the hour ending at 8 a. m. however, 1470 gal. is the estimated requirement, 670 gal. less than the quantity heated. This being drawn from storage, reduces the storage of hot water to 330 gal.

The estimated draft for the hour following is 800, which being the amount heated, the storage would remain the same—330 gal. The estimated requirement in the hour ending at 10 a. m. being 100 gal. and the amount heated being 800 gal.; 700 gal. of hot water would be added to the

Form No. 101

APARTMENT HOUSE
150 PEOPLE
40 APARTMENTS

HOT WATER ESTIMATE SHEET

#2

These ratings are based on hot water at 180° F

	Showers St'd 4" head 15 gal per bath-8 min	40 Bath tubs 15 gal per bath	40 Kitchen taps 5 gal. per hour	2 Slop sink taps 15 gal. per hour	40 Hand lavatories 1 gal per service	12 Set Tubs 20 Gals. Ea.					Pounds of Steam used per hour	Total required Gals. per hour	Gallons heated per hour	Hourly surplus Gallons	Total surplus Gallons
1 A. M.	MAXIMUM HOURLY STEAM REQUIRED = 480 LBS = 16 HP = 1920 ☐' RADIATION														
2	FULL STEAM LOAD SUSTAINED 7 HOURS OUT OF 18 COVERED BY ESTIMATE														
3	= AVERAGE STEAM LOAD OF 317 LBS STEAM PER HOUR = 66% OF MAX.														
4	STEAM LOAD														
5				STORAGE CAPACITY 180% OF HOURLY HEATING											
6		80	50		10						156	140		+260	720
7		250	100		50						480	400		+ 0	720
8		450	150	30	100	100					480	830		430	290'
9		150	100	30	50	100					480	430		- 30	260
10				30	20	100					180	150	400 GALS PER HOUR	+250	510
11			100	30	20	100					300	250		+150	660
12 M.			170	30	100	100					480	400		+ 0	660
1 P M			150		100						300	250		+150	720
2			50								60	50		+350	720
3											0	0		+400	720
4		150									180	150		+250	720
5		300	50								420	350		+ 50	720
6		150	150		100						480	400		+ 0	720
7			200		100						360	300		+100	720
8		.	100		100						240	200		+200	720
9		150									180	150		+250	720
10		600									480	600		-200	520
11		450									480	450		-50	470
12	Steam Pressure											0		+400	720

Steam available from HEATING SYSTEM ½ LB. PRESSURE

Water Pressure

storage, thus bringing the total of storage back to 1000 gal., the maximum provided for. In the following hours of the day until 10 p. m. no big water demand exists and the amount heated remains the same. During that hour 900 gal. is the estimated draft which exceeds the amount heated by 100 gal., thus reducing the storage capacity to 900 gal. This is graphically shown on Chart 1, the full lines at left of sheet indicating the drafts from different fixtures; the heavy dotted line, the total hot water draft; the line of crosses, the steam requirement, and the line of small circles the net storage of hot water. This is an inefficient installation, the steam demands fluctuating and the storage capacity remaining constant the greater part of the day. This installation was improved in service by increasing the storage capacity and cutting out of service a portion of the heating surface.

Estimate 2 covers an apartment house of forty apartments, housing 150 people. In this the peak demand occurs, as in a hotel, in the early morning and late at night with a peak at noon of short duration. In

APARTMENT HOUSE
130 PEOPLE
93 APARTMENTS

HOT WATER ESTIMATE SHEET

Form No. 101
#3A

These ratings are based on hot water at 180° F

	Showers 8i'd 6" head 10 gal. per bath-5 min.	93 Bath tubs 20 gal. per bath	11 Kitchen taps 15 gal per hour	8 Slop sink taps 15 gal per hour	99 Hand lavatories 1 gal per service	1 DISH WASHER					Pounds of Steam used per hour	Total required Gals. per hour	Gallons heated per hour	Hourly surplus Gallons	Total surplus Gallons
1 A. M.	MAXIMUM HOURLY STEAM REQUIRED = 480 LBS = 16 HP = 1920 [□'] RADIATION														
2	FULL STEAM LOAD SUSTAINED 10 HOURS OUT OF 18 COVERED BY ESTIMATE														
3	= AVERAGE STEAM LOAD 360 LBS PER HOUR = 75% OF MAX LOAD														
4	4 HOURS CONTINUOUS LOAD, 3 HOURS CONTINUOUS LOAD														
5	STORAGE CAPACITY 305% OF HOURLY HEATING														
6		200	100		50						420	350		+50	1220
7		600	100		100						480	800		-400	820
8		600	100	50	100	150					480	1000		-600	220
9		200		100	50	200					480	550	400 GALS. HEATED PER HOUR	-150	70
10				100							120	100		+300	370
11			100	100							240	200		+200	570
12 M.			100	100	100						360	300		+100	670
1 P M			100		100	100					360	300		+100	770
2			100	100	100	200					480	500		-100	670
3				100							120	100		+300	970
4				100							120	100		+300	1220
5		200	100	100							480	400		+000	1220
6		600	100	100	100						480	900		-500	720
7		200	100		100	100					480	500		-100	620
8			100		100	300					480	500		-100	520
9						100					120	100		+400	920
10		400									480	400		+000	920
11		600									480	600		-200	720

12 Steam Pressure

Steam available from HEATING SYSTEM

Water Pressure

this particular project two estimates were made, one covering an hourly heating of 400 gal. and a storage of 720 gal. and the other covering an hourly heating of 800 gal. and a storage of 400 gal. With each quotation was included the steam boiler necessary for steam requirement and the big steam water heater with small steam boiler totalled about 2/3 in price that of the small steam water heater with big heating unit and big steam boiler required to supply the necessary steam. With small hourly heating a small fire would be maintained for 24 hr. to provide for three peaks, morning, noon and night, instead of a big fire to render the same service.

By looking at the chart, it will be seen that the peak drafts occur in hours ending at 7, 8 and 9 a. m. at the end of which period the storage becomes reduced from 720 gal. to 620 gal. with a fairly well sustained demand on the heater until noon, which does not recover full storage capacity until 1 p. m. and remains at full storage of hot water until the hour ending at 10 p. m., when it is reduced to 520 gal. and at 11 o'clock

APARTMENT HOUSE
130 PEOPLE
93 APARTMENTS

HOT WATER ESTIMATE SHEET

Form No. 101
#3B

These ratings are based on hot water at 180° F

	Showers Std 4" head 10 gal per bath-5 min.	93 Bath tubs 20 gal per bath	11 Kitchen taps 5 gal per hour	8 Slop sink taps 15 gal per hour	99 Hand lavatories 1 gal per service	1 DISH WASHER					Pounds of Steam used per hour	Total required Gals. per hour	Gallons heated per hour	Hourly surplus Gallons	Total surplus Gallons
1 A. M.	MAXIMUM HOURLY STEAM REQUIRED = 960 LBS = 32 HP = 3840 □' RADIATION														
2	FULL STEAM LOAD SUSTAINED 3 HOURS OUT OF 18 COVERED BY ESTIMATE														
3	= AVERAGE LOAD OF 491 LBS. STEAM PER HOUR = 51% OF MAX LOAD														
4	STORAGE CAPACITY = 51 % OF HOURLY HEATING														
5															
6		200	100		50						420	350		+450	420
7		600	100		100						960	800		+ 0	420
8		500	100	50	100	150					960	900		-100	320
9		200		100	50	200					660	550		+250	420
10				100							120	100	800 GALS. PER HOUR	+700	420
11			100	100							240	200		+600	420
12 M			100	100	100						360	300		+500	420
1 P. M.			100		100	100					360	300		+500	420
2			100	100	100	200					600	500		+300	420
3				100							120	100		+700	420
4				100							120	100		+700	420
5		200	100	100							480	400		+400	420
6		600	100	100	100						960	900		-100	320
7		200	100		100	100					600	500		+300	420
8			100		100	300					600	500		+500	420
9						100					120	100		+700	420
10		400									480	400		+400	420
11		600									720	600		+200	420

12 Steam Pressure

Steam available from HEATING SYSTEM

Water Pressure

to 470 gal. with ample time for recovery before the peak demand of early morning.

Sheet No. 3A and 3B cover apartment houses with more than twice as many bath tubs and occupied by 130 people. The estimates for each hour are made in a manner similar to the other and as in the other estimates, the peak demand exists from 6 to 9 a. m. with a short peak between 1 and 2 o'clock and an extended peak from 4 to 8 o'clock, a shorter peak from 10 to 11 o'clock. Assuming an hourly heating capacity of 400 gal. and a storage capacity of 305 per cent or 1220 gal., at the end of the morning peak, there would remain in hot water storage only 70 gal. and full storage capacity would not be recovered until 4 p. m. The late afternoon draft reduces storage capacity to 520 gal. and the night peak demand reduces amount stored to 720 gal. with several hours in which the water may be heated slowly to recover full storage capacity for the early morning peak.

Chart 3A shows in line of crosses, the steam requirement running fairly uniform throughout the day with a maximum of 480 lb. of steam,

Y.W.C.A HOT WATER ESTIMATE SHEET Form No. 101 #4A

These ratings are based on hot water at 180° F.

	Showers St'd 4" head 20 gal. per bath-5 min. 17	Bath tubs 20 gal per bath 17	Kitchen taps 5 gal per hour 13	Slop sink taps 4½ gal per hour 5	Hand lavatories 1 gal per service 52	8 HAND TUBS	1 DISH WASHER			Pounds of Steam used per hour	Total required Gals per hour	Gallons heated per hour	Hourly surplus Gallons	Total surplus Gallons
1 A. M.	MAXIMUM HOURLY STEAM REQUIRED = 720 LBS = 24 HP = 2880 ☐' RADIATION													
2	FULL STEAM LOAD SUSTAINED 10 HOURS OUT OF 18 COVERED BY ESTIMATE:													
3	= AVERAGE LOAD 570 LBS. STEAM PER HOUR = 79% OF MAX LOAD													
4	MAXIMUM LOAD SUSTAINED FOR 7 HOURS CONTINUOUSLY													
5	STORAGE CAPACITY = 475% OF HOURLY HEATING													
6	100	100	50		30	50				396	330		+270	2850
7	200	200	50		100	50				720	600		+ 0	2850
8	300	300	100	50	200	50	100			720	1100		-500	2350
9	100	100	100	50	100	50	100			720	600		+ 0	2350
10		40	50	50			100			128	240	600 GALS PER HOUR	+360	2760
11			50	50						120	100		+500	2850
12 M.			100	50	50					240	200		+400	2850
1 P. M.			100		100		100			360	300		+300	2850
2			100	50	50		100			360	300		+300	2850
3			50	50			100			240	200		+400	2850
4	800		50	50						720	900		-300	2550
5	1000	40	50	50						720	1140		-540	2010
6	600	100	100	50	50					720	900		-300	1710
7	400	100	100		100		50			720	750		-250	1460
8	600	100	100		50	50	100			720	1000		-400	1060
9	1000	200	50			100	100			720	1450		-850	210
10	400	200				100	50			720	750		-150	60
11	100	100								240	200		+400	460

12 Steam Pressure 0 TO 2 LBS

Steam available from HEATING SYSTEM WITH VACUUM PUMP

Water Pressure

equal to 16 hp., the equivalent of 1920 sq. ft. of radiation. On the other hand, the storage of hot water fluctuating from 1220 to 70 gal. and remaining constant for a brief period.

Sheet 3B shows an estimate made for the same apartment in which it is assumed that the hourly heating capacity will be double that of the previous estimate and the storage capacity will be 51 per cent of the hourly heating. In this estimate, the hourly hot water demand exceeds the amount heating in one hour only from 7 to 8 a. m. during which the storage capacity of hot water is reduced from 420 to 220 gal.

Chart 3B covering Estimate 3B shows a fluctuating steam requirement of 900 lb., 32 hp., equal to 3840 sq. ft. of radiation or double that covered in estimate and Chart 3A. The storage remains very nearly constant throughout the day. With the latter proportion of heater, doubtless twice as much fuel would be burned to secure the same result as with former, and the initial cost would probably be double.

YW.C.A. **HOT WATER ESTIMATE SHEET.** Form No. 101 #4B

These ratings are based on hot water at 180° F.

	Showers Std 4" head 20 gal. per bath-5 min. 17	Bath tubs 20 gal per bath 17	Kitchen taps 5 gal. per hour 13	Slop sink taps 5 gal. per hour 5	Hand lavatories 1 gal per service 1	8 Hand Tubs	1 Dish Washer				Pounds of Steam used per hour	Total required Gals per hour	Gallons heated per hour	Hourly surplus Gallons	Total surplus Gallons
1 A. M.	Maximum Hourly Steam Required = 1200 Lbs = 40 HP = 4800 ☐' Radiation														
2	Full Steam Load Sustained 4 Hours Out Of 18 Covered By Estimate														
3	= Average Load 695 Lbs Steam Per Hour = 58% Of Maximum Load														
4	Maximum Load Sustained For 2 Hours Continuously Heating														
5	Storage Capacity = 52½ % Of Hourly Heating														
6	100	100	50		30	50					396	330		+670	525
7	200	200	50		100	50					720	600		+400	525
8	300	300	100	50	200	50	100				1200	1100		-100	425
9	100	100	100	50	100	50	100				720	600		+400	525
10		40	50	50			100				230	190	1000 Gals. Per Hour	+810	525
11			50	50							120	100		+900	525
12 M.			100	50	50						240	200		+800	525
1 P. M.			100		100		100				360	300		+700	525
2			100	50	50		100				360	300		+700	525
3			50	50			100				240	200		+800	525
4	800		50	50							1080	900		+100	525
5	1000	40	50	50	50						1200	1150		-150	375
6	600	100	100	50	100						1140	950		+50	425
7	400	100	100		50		50				960	800		+200	525
8	600	100	100			50	100				1200	1000		+000	525
9	1000	200	50			100	100				1200	1400		-400	125
10	400	200				100	50				900	750		+250	375
11	100	100									240	200		+800	525
12	Steam Pressure 0 To 2 Lbs.														

Steam available from Heating System With Vacuum Pump

Water Pressure

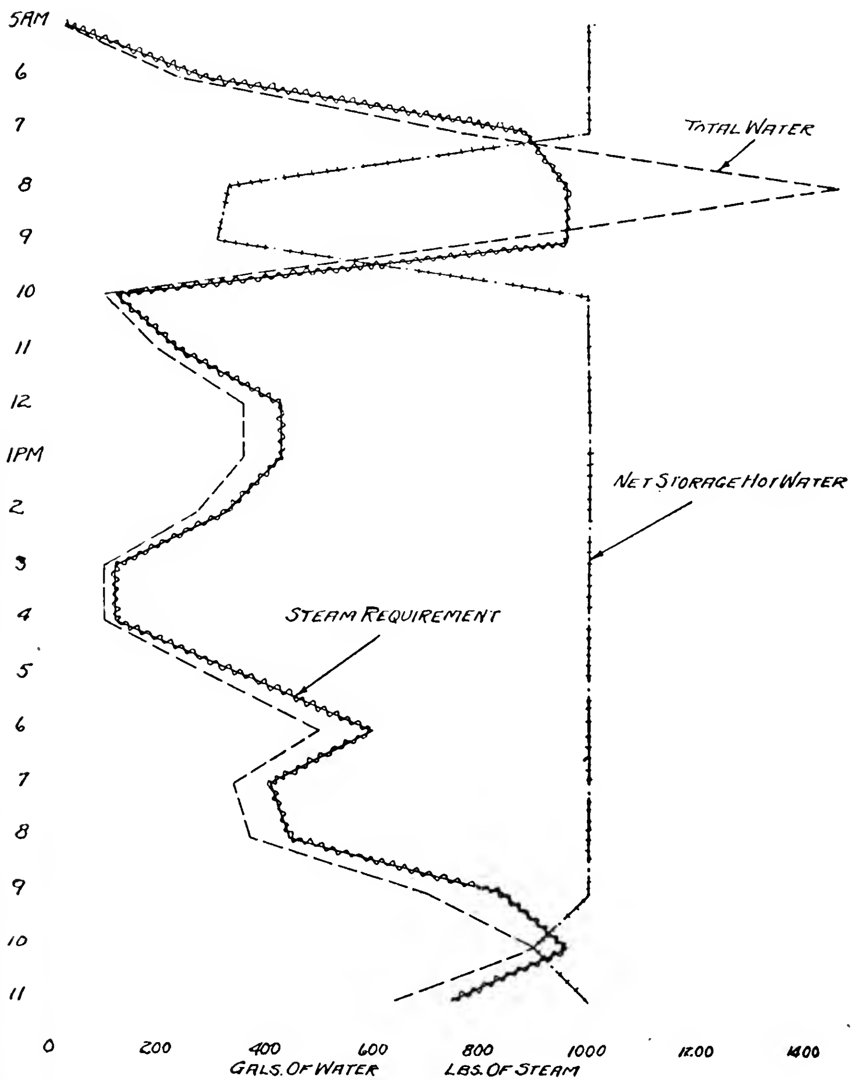
Sheet 4A and 4B cover the requirement for a Y. W. C. A. The dormitory draft will produce a peak demand in the early morning and there will be a fairly sustained demand from 3 o'clock until 10 or 11 p. m. In Sheet 4A we have estimated 600 gal. per hr., equalling a maximum steam requirement of 720 lb. equal to 24 hp., the equivalent of 2080 sq. ft. of radiation. With this small hourly demand we provided a storage capacity of 475 per cent or 2850 gal.

This large storage is reduced in the morning peaks from 7 to 9 o'clock to 2350 gal. and in the extended peak from 3 to 10 p. m. is reduced to

CHART No. 1
300 ROOM HOTEL

INEFFICIENT IN SERVICE
STEAM REQUIREMENT NOT SUSTAINED
STORAGE ADEQUATE
STORAGE = 125% HOURLY HEATING

= STEAM REQUIREMENT., = NET STORAGE HOT WATER; ——— = TOTAL WATER.



100 gal. In the chart the line of crosses indicating steam demand shows a fairly uniform load throughout the day with a fluctuating storage.

Estimate 4B covers the same project in which an hourly heating capacity of 1000 gal. is estimated with a storage capacity of 52½ per cent. The early morning peak reduces storage capacity from 525 to 425 gal. and the extended peak at night reduces storage capacity at 9 p. m. to 125 gal. In this case there is a maximum hourly steam requirement of 1200 lb. equal to 40 hp. equivalent to a demand of 4800 sq. ft. of radiation and a big fire must be maintained for 24 hr. to cover a peak requirement of one hour in the morning and one hour in the afternoon and another hour at night, which equal or exceed the heating capacity of the steam plant.

Chart 4B shows the uneven steam requirement in line of crosses and the almost constant storage capacity.

I will admit that the comparative sheets and charts show extremes which are intended to fortify my contention that hot water storage performs the same functions in hot water supply as does the fly wheel in the delivery of power from an engine. With storage for heat provided in coal bin, in fire chamber, in steam dome on boiler, and in steam water heater the most economical place for storage of heat is in the steam water heater, if quick response to demand is the primary consideration.

CHART No. 2
APARTMENT HOUSE, 150 PEOPLE, 40 APARTMENTS
EFFICIENT IN SERVICE
SMALL PEAK STEAM REQUIREMENT EVENLY SUSTAINED
STORAGE ADEQUATE FOR PEAKS
STORAGE = 180% HOURLY HEATING
= STEAM REQUIREMENT, = NET STORAGE HOT WATER, - - - = TOTAL WATER

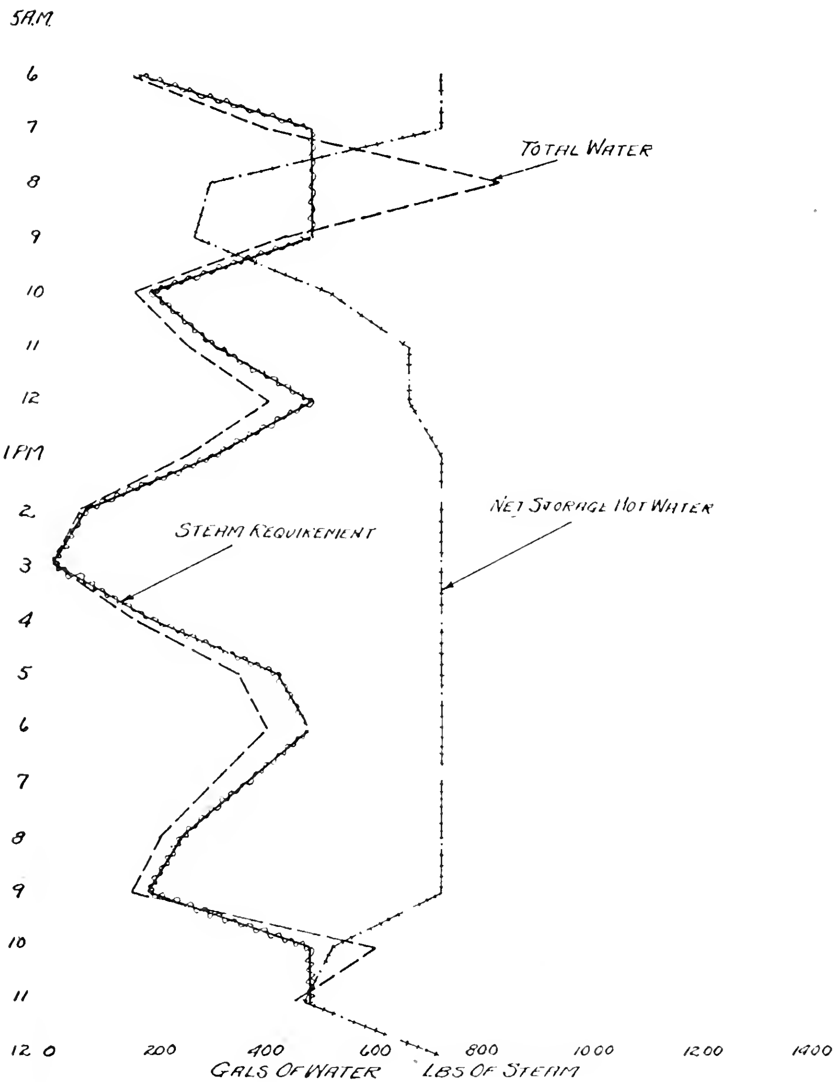


CHART No. 3A
APARTMENT HOUSE, 130 PEOPLE, 93 APARTMENTS.
EFFICIENT IN SERVICE
SMALL PEAK STEAM REQUIREMENT EVENLY SUSTAINED
STORAGE ADEQUATE FOR PEAKS
STORAGE = 305% HOURLY HEATING
=STEAM REQUIREMENT, =NET STORAGE HOT WATER --- =TOTAL WATER

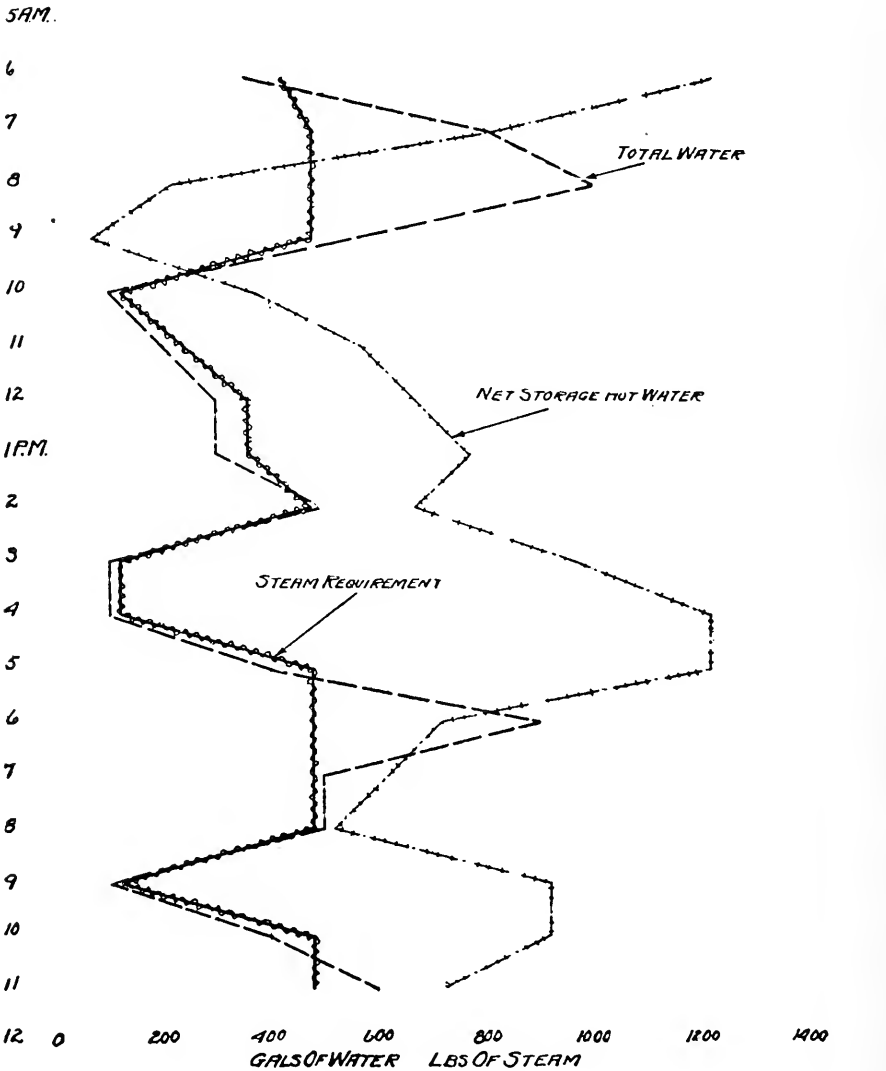


CHART Nº 3 B
APARTMENT HOUSE, 130 PEOPLE, 93 APARTMENTS
INEFFICIENT IN SERVICE
BIG PEAK STEAM REQUIREMENT NOT SUSTAINED
STORAGE REDUCED BUT LITTLE FOR PEAK DRAFTS
STORAGE = 51% HOURLY HEATING

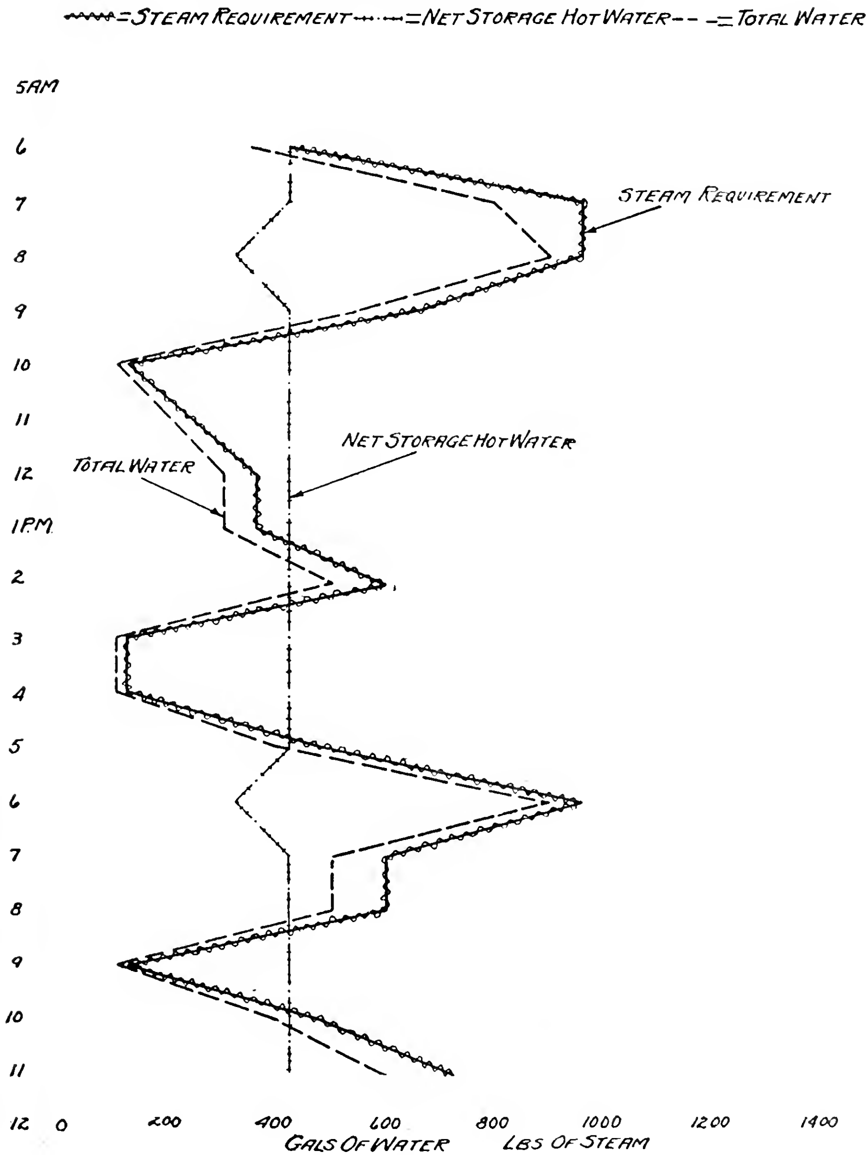
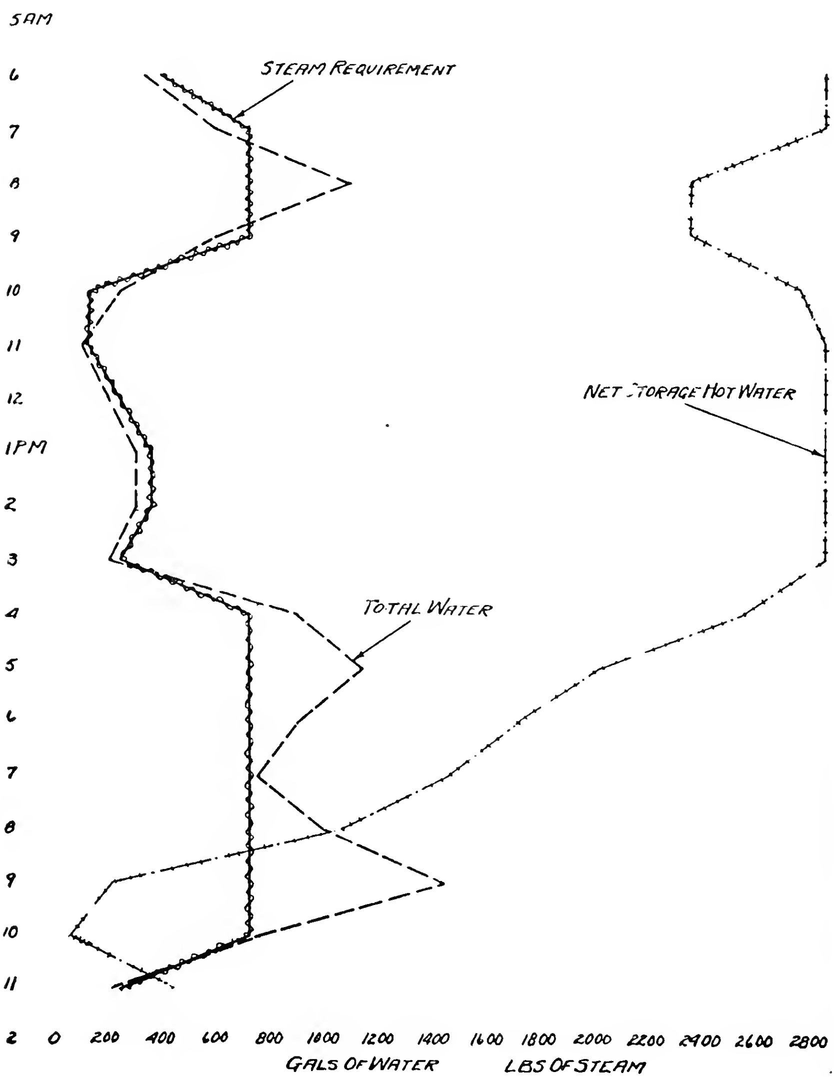


CHART No 4 A
YWCA
EFFICIENT IN SERVICE
SMALL STEAM REQUIREMENT FAIRLY EVENLY SUSTAINED
STORAGE ADEQUATE FOR PEAKS
STORAGE = 475% HOURLY HEATING
=STEAM REQUIREMENT —·— =NET STORAGE HOT WATER - - - =TOTAL WATER



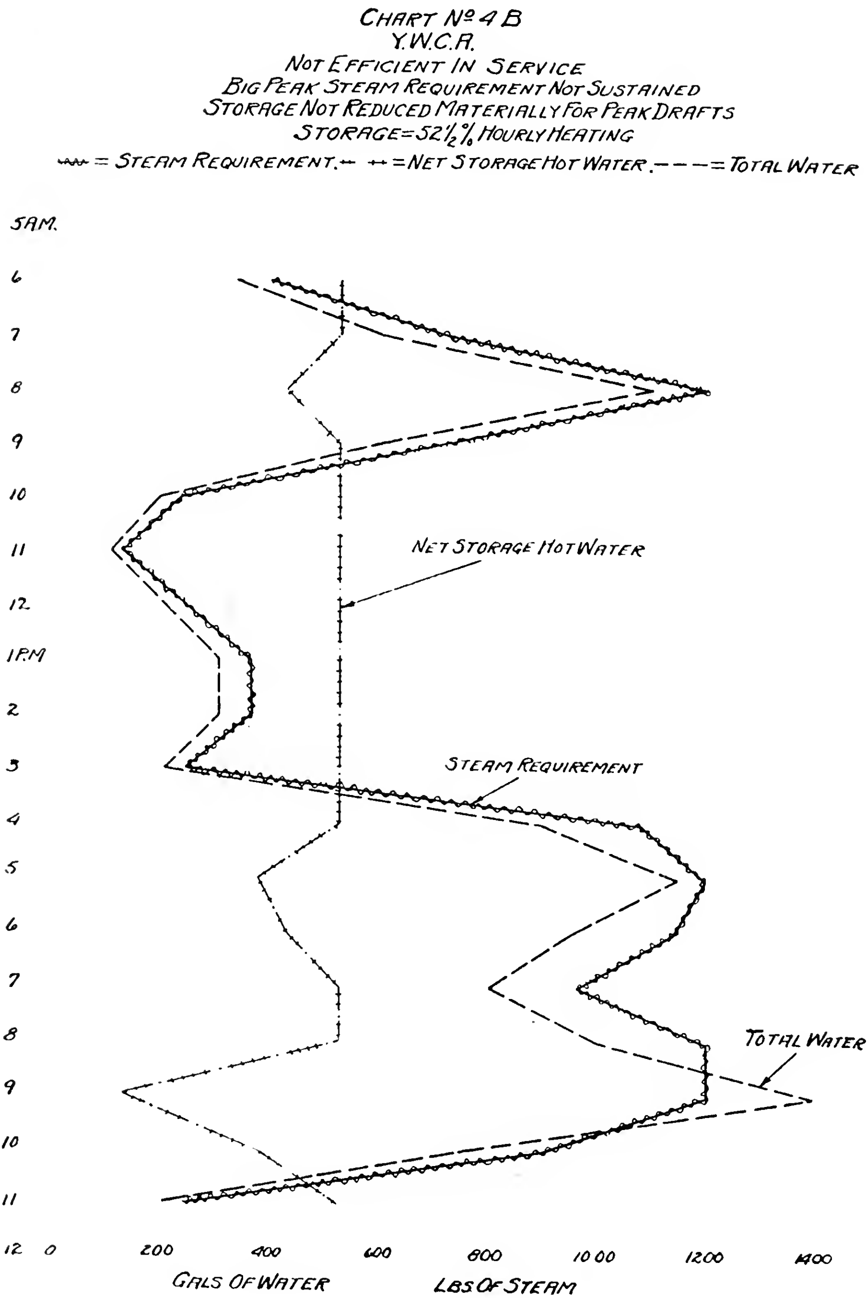


Chart No 4 B
Y.W.C.A.
Not Efficient In Service
Big Peak Steam Requirement Not Sustained
Storage Not Reduced Materially For Peak Drafts
Storage = 52½% Hourly Heating
= Steam Requirement. = Net Storage Hot Water. ---- = Total Water
5 A.M.
6
7
8
9
10
11
12
1 P.M
2
3
4
5
6
7
8
9
10
11
12
0
200
400
600
800
1000
1200
1400
Net Storage Hot Water
Steam Requirement
Total Water
Gals Of Water
Lbs Of Steam

No. 596

DESIGN OF LARGE BOILER PLANTS

By J. Grady Rollow[1], Wilmington, Del.

Non-Member

THE questions which an engineer must determine before he can successfully design a boiler plant are:

1. Character of the fuel to be used.
2. Location of the plant with respect to fuel supply.
3. Character of the water to be used as boiler feed.
4. Size and character of the load to be carried.
5. Distribution and use of steam.
6. Cost of labor, class available and location of plant with respect to labor markets.
7. Cost of fuel.
8. Climatic conditions and altitude.
9. Importance of the time element in getting plant either partially or wholly in operation.
10. Local laws governing design and manufacture of pressure vessels, issuance of smoke, workmen's compensation acts, etc.
11. The return expected on the capital invested.
12. Permanence of plant.
13. Value of real estate.

The first item for consideration is the fuel supply. Fuels occur in all the three states of matter, viz.: solid, liquid, and gaseous. At the present time, nearly all the large plants burning gas are those in connection with blast furnaces, although there are still a few in West Virginia, Ohio and California burning natural gas. Ambrose N. Diehl has reported[2] experiments with four distinct types of burners applied to boilers with several modifications of each. The conclusions reached from a study of Mr. Diehl's work is that high boiler efficiency is not absolutely dependent upon the design of the burner. With a properly designed furnace, clean boiler, tight setting and baffles, and careful regulation, high efficiencies can be gotten with the common-type burner having a narrow orifice at the exit and no provision for mixing the gas and air within the burner.

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[1] Combustion Engineer, E. I. duPont de Nemours & Co., Wilmington, Del.

[2] Burning Blast Furnace Gas, published in the 1915 Year Book of the American Iron and Steel Institute.

Paper presented originally before Eastern Pennsylvania Chapter; also at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1921.

A furnace of ample length with an incandescent roof is most essential. The experiments clearly showed that a scientifically designed burner was a great aid in obtaining good regulation, and with the class of help ordinarily found in boiler plants, would give better results than common burners. Some modification of the Birkholz-Terbeck burner is considered by a number of eminent engineers to best meet the requirements. This burner works equally satisfactory on producer gas. Its success is no doubt due, in a large measure, to the fact that it is fitted with an observation glass concentric with its axis so that the operator can see his flame, as much as to the flexibility of the air control. Another burner which has attracted considerable attention is the Bradshaw burner which is of the aspirating type based on the venturi principle. As gas pressures are constantly changing in blast-furnace work requiring constant air adjustments for proper combustion, this burner was designed to automatically draw in its own air supply. The draft in the furnace is automatically held at a predetermined pressure near zero. When the gas supply increases, due to increase in gas pressure, the velocity at the throat increases, drawing in a larger air supply. Very good results have been obtained with this burner at the Cambria steel plant at Johnstown, Pa., and the Pittsburgh Steel Co., Monessen, Pa.

An average blast furnace gas on lake ores runs: CO_2, 12.5 per cent; CO, 25.4 per cent; H, 3.5 per cent; N, 58.6 per cent; and at a temperature of 62 deg. fahr. and 30 in. barometer contains 92 B.t.u. per cu. ft. The heat value of blast furnace gas varies from 85 to 100 B.t.u. per cu. ft. under standard conditions. Ordinarily, it requires approximately a net ton of coke per gross ton of pig iron produced. From 45 to 50 per cent of the heat value of the fuel used in a blast furnace passes off at the top. Of this, about 10 per cent is lost, 30 per cent burned in the hot blast stones and the remaining 60 per cent used for producing power either under steam boilers or in gas engines.

The Babcock & Wilcox Co. states that 0.8 sq. in. of gas opening per rated hp. will enable a boiler to develop rating with 2 in. gas pressure. As gas pressures run from 6 to 8 in., the above proportion will be good for ordinary overloads. The air openings should also be about 0.8 sq. in. per rated hp. Boilers burning blast-furnace gas should be provided with approximately 0.75 in. of draft at the damper. It is safe to figure 150 per cent of rating on a boiler properly set and cared for when using blast-furnace gas.

The subject of burning liquid fuel has been so ably presented to the Society by Mr. Peabody that no further comment is necessary here.

There are a number of plants in the United States using waste wood, either as part or all of their fuel supply. Wood burning requires large furnace volumes. Just how far can be successfully gone in the matter of large furnaces for dry waste wood is not known, but the illustration of the furnaces for the St. Louis Manufacturing Co. will show that some long steps have been taken. This plant has been in successful operation for six months and ratings of 200 per cent have been developed for 24 hr. The fuel there is dry sawdust and shavings. •

The Georgetown plant of the E. I. duPont de Nemours & Co. which manufactures grain or ethyl alcohol from wood waste from the Atlantic

Coast Lumber Co., burns about the lowest grade wood waste on record. The material is 50 to 80 per cent sawdust and the rest, hogged chips.

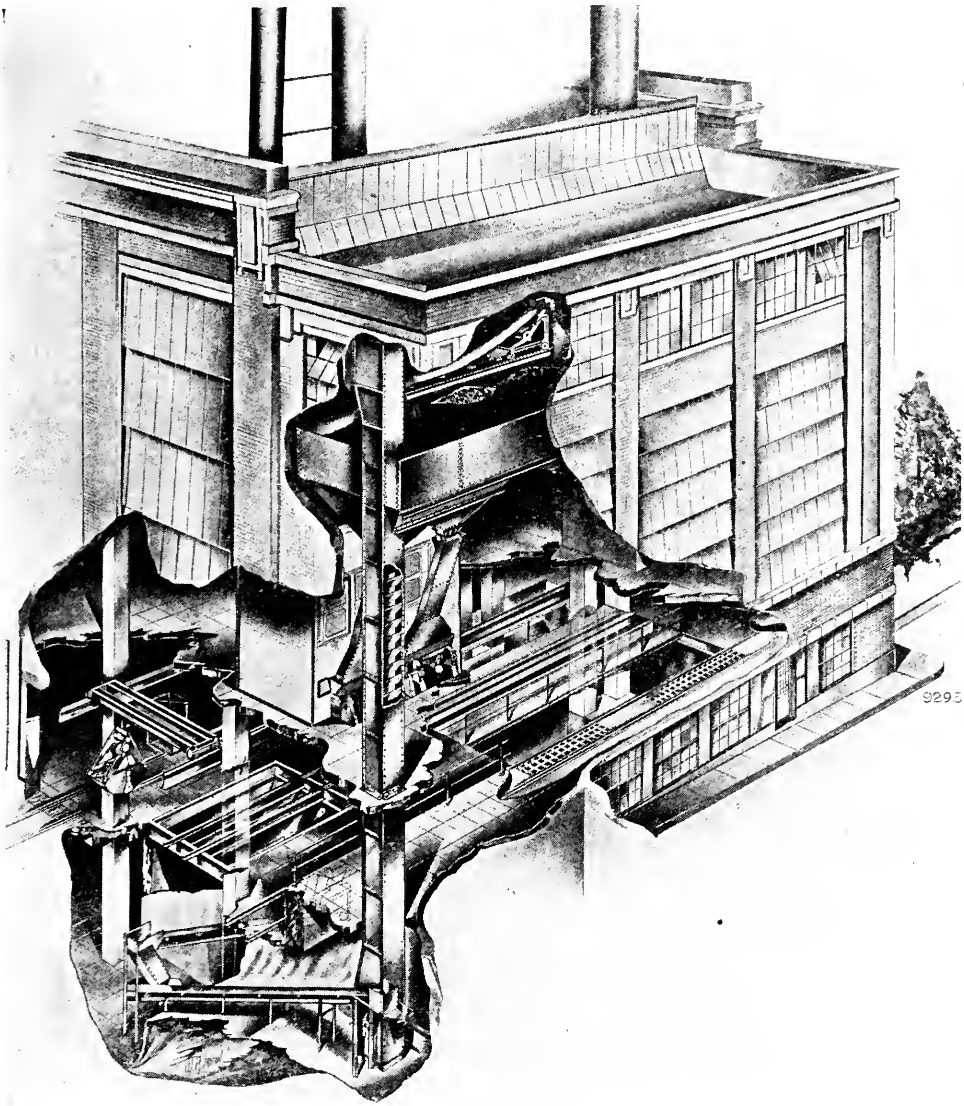

FIG. 1. ILLUSTRATION OF COAL-HANDLING EQUIPMENT OF A CENTRAL STATION WITH OVERHEAD BUNKER AND STORAGE IN SUB-BASEMENT

After going through a digesting and liquor extracting process, it is run through an Edgerton garbage press which is essentially an apron feeder moving between the rolls of a glorified clothes wringer. After going

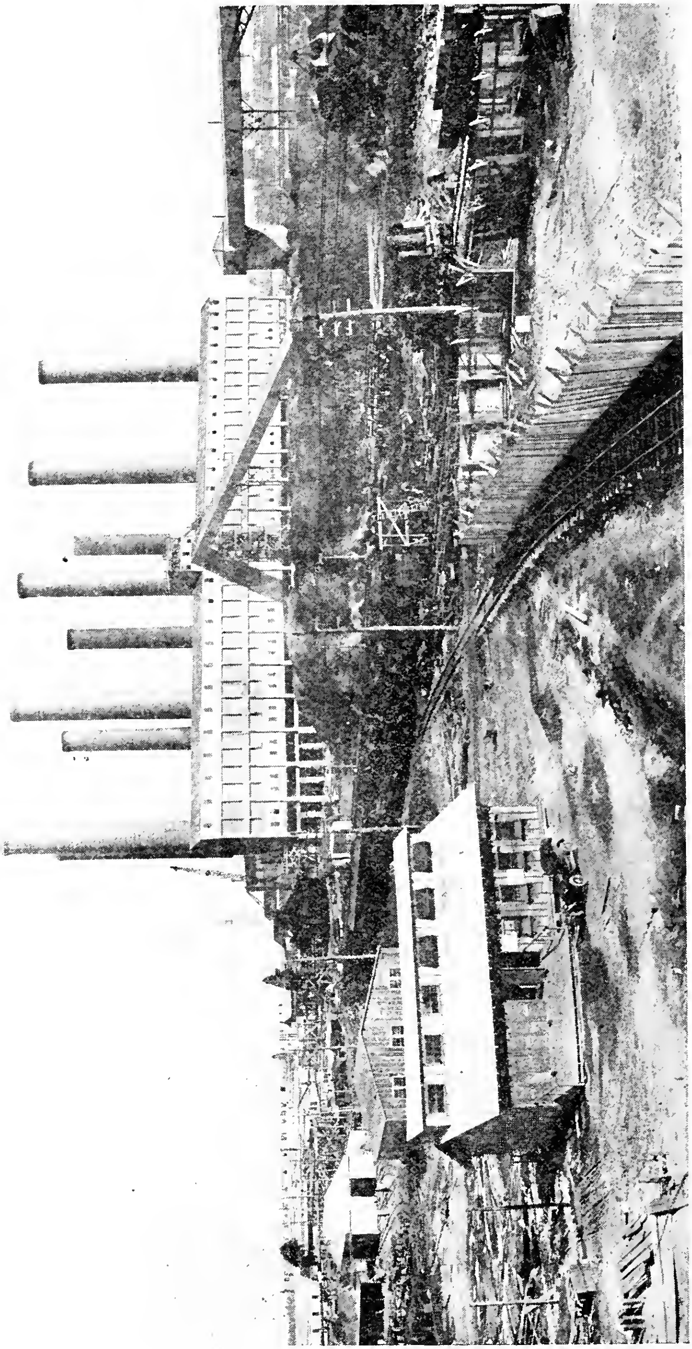

FIG. 2. PANORAMIC VIEW OF OLD HICKORY BOILER HOUSE

through this dehydrating process, the spent wood containing 60 per cent moisture is delivered to the boiler furnaces by conveyors. The heat value of this material is 9100 B.t.u. per lb. dry and this plant burns sufficient of this material to develop rating on the boilers. Continuous ratings of as high as 130 per cent may be obtained with wood waste running 50 per cent or less in moisture. Essential features of a well designed furnace for wet wood waste are:

1. Restriction at the furnace outlet;
2. Combustion chamber between furnace outlet and the innermost pile.

There are some comparatively large boiler plants operating in the cane-sugar industry which burn green bagasse. This fuel has a net heat value of from 2200 to 3350 B.t.u. per lb. In the early days of the industry in this country when labor was cheap and European competition played no part, the very crudest methods were used in the extraction of the juice. Oftentimes the extraction was sacrificed so that there would be enough bagasse for fuel, and the bagasse was sun-dried before burning. Today with modern methods of manufacture, the extraction has reached 80 per cent and above, and the bagasse can be burned without drying. Naturally the higher the extraction, the better the grade of fuel that is produced, but at the same time, the more juice there is to be evaporated, the more power consequently required per ton of cane, so that today the problem of the engineer in the cane-sugar industry, is to design a plant sufficiently economical to generate the power with the bagasse. Professor Kerr has found out that the bagasse from one ton of cane will produce from 1.16 to 1.44 boiler hp. for 24 hr.

At present it is necessary to use auxiliary fuel in the United States and the West Indies, although it is the belief of some engineers that the time will come when bagasse will supply the entire fuel for that industry.

The Babcock & Wilcox Co. have developed a furnace for burning green bagasse which pretty nearly meets the requirement. It has been found that bagasse burns better on a hearth than on grates, the air being supplied through a series of tuyeres by natural draft. Large combustion space is necessary and it is more economical to burn it in large quantities. Thus a furnace has been developed which will serve two boilers. With 0.5 in. draft in this furnace, 450 lb. of 50 per cent moisture bagasse per sq. ft. of hearth per hr. can be burned; 300 lb. per sq. ft. per hr. however gives the best results. An essential feature of a well-designed bagasse furnace is provision for catching and removing the dust which will fuse into a grass-like clinker if allowed to accumulate. The fuel is introduced into the top of the furnace by conveyors which should be so designed as to give a constant fuel feed with a minimum admission of air.

Bagasse-burning boiler plants usually give from 90 to 100 per cent of rated capacity. There is no particular reason why greater capacities cannot be obtained as it is just a question of how much fuel the furnace can burn.

The most important fuel for the generation of steam in stationary power plants is coal. It is more widely distributed than any fuel (wood excepted), is found in much larger quantities, varies more widely in characteristics, and has demanded more attention from engineers than all other fuels combined. Coals are classified as follows:

Kind	Fixed Carbon, per cent	Volatile, per cent
Anthracite	97. to 92	3.0 to 7.5
Semi-anthracite	92.5 to 87.5	7.5 to 12.5
Semi-bituminous	87.5 to 75	12.5 to 25.
Bituminous (or Eastern)	75 to 60	25.0 to 40.
Sub-bituminous (or Western bituminous)	65 to 50	35. to 50
Lignite	Under 50	Over 50

Anthracite which is geologically the oldest, is found in eastern Pennsylvania, in Pulaski and Wythe counties of Virginia, Gunnison County, Colorado, and in Santa Fe and Socora Counties of New Mexico. The only anthracite fields of importance, however, are the five of Eastern Pennsylvania which are enumerated as follows in *Mineral Resources* for 1886:

(1). Southern or Pottsville field.
(2). Western or Mahoney and Shamokin field.
(3). Eastern middle or upper Lehigh field.
(4). Northern or Wyoming and Lackawanna fields.
(5). Loyalsock and Mehoopany fields.

Anthracite is the easiest of all coals to fire by hand and the hardest to handle successfully with stokers. In this connection the following bit of ancient history may be of interest to Philadelphians:

"In 1793, The Lehigh Coal Mining Co. was formed, which some years later sold a quantity to the City of Philadelphia for the use of a steam engine at the water works, then at Broad and Market Streets, but it was not used because it could not be burned."

"In 1812, Col. George Shoemaker took nine wagon loads to Philadelphia, disposed of two or three loads at the cost of handling and left the rest with different persons for experiment. At the Fairmount Wire & Nail Works, the workmen spent a forenoon in fruitless attempts to make a fire with it. At last they closed the furnace doors and went to dinner; returning an hour later they found the doors red hot and the furnace aglow."

"In 1820 the trade was fully established, 365 tons being shipped to Philadelphia that year."—WILLIAM KENT in *Steam Boiler Economy*.

Today the demand for anthracite coal for domestic purposes is so keen that engineers of steam plants need to concern themselves with the buckwheat sizes and smaller only. Sizes smaller than No. 3 buckwheat cannot be burned successfully in hand-fired furnaces unless mixed with some bituminous coal. The Arlington Plant of the duPont Co. successfully burns a mixture of one part bituminous to two parts of small barley, obtaining efficiencies of 68 per cent and ratings of 140 per cent. During the recent soft-coal strike, a mixture of one part of bituminous to four of anthracite was used, but with a decrease in capacity of 110 per cent of rating and a corresponding decrease in efficiency. The steam sizes of anthracite require forced draft. Fans should be provided which will give 3 in. of static pressure at the fan where hand firing is used. The common practice in the anthracite region a few years ago was to use stacks from 25 to 50 ft. above the boiler setting. This is entirely wrong as it limits the capacity very materially. There is less reason why anthracite with forced draft should require short stacks than underfeed-stoker plants burning bituminous, because anthracite requires higher drafts for the same combustion rates.

The best known of the early types of mechanical stokers for burning anthracite coal is the Wilkenson. In recent years stokers of the chain-grate type using forced draft have been developed for burning small sizes of anthracite. This type of stoker will also burn coke braize or bituminous coal. It will burn continuously 25 lb. of No. 3 buckwheat anthracite coal per sq. ft. of grate per hr. An essential feature of a

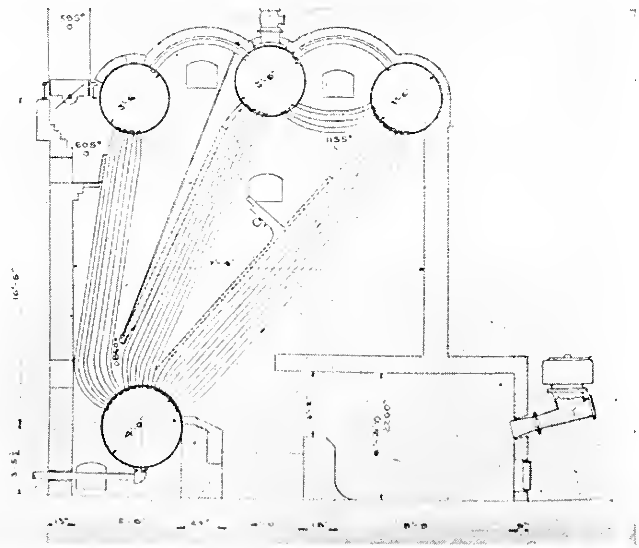

FIG. 3. FURNACE FOR BLAST-FURNACE GAS USING MODIFICATION OF BIRKHOLTZ-TERBECK BURNER

stoker for this work is the dividing of the wind box into three or more compartments so that different pressures may be carried in different zones.

Semi-anthracite coals are found in Sullivan County, Pa., and in smaller amounts in the Dan River field of North Carolina and Virginia and the Deep River field of North Carolina. The total amount of semi-anthracite coal is insignificant. The furnaces which handle anthracite satisfactorily will also burn semi-anthracite.

Semi-bituminous coals form a very important group, which includes the famous Pocahontas and New River fields. Coals belonging to this class are found in Center, Huntingdon, Cambria, Clearfield, and Somerset Counties of Pennsylvania, Allegheny and Garret Counties of Maryland, Tazewell County of Virginia, Fayette, McDowell, Mercer, Mineral Raleigh and Tucker Counties of West Virginia, Franklin, Johnson, Logan and Sebastian Counties of Arkansas, and Walker County of Georgia. Fuels belonging to this group, require more furnace volume than the anthracites and semi-anthracite. Stokers of the underfeed type handle them very successfully. The inclined-grate overfeed types will also handle them.

Bituminous coals are found in Alabama, Arkansas, Colorado, Illinois, Indiana, Iowa, Kansas, Kentucky, Michigan, Missouri, Montana, New Mexico, Ohio, Oklahoma, Pennsylvania, Tennessee, Texas, Utah, Virginia, Washington, West Virginia, and Wyoming. The bituminous coals offer more opportunities for the various types of stokers. The better grades of bituminous can be successfully burned in the underfeed, and the inclined-grate overfeeds. The chain-grate handles the low-grade bituminous coals very successfully, although the underfeeds have been invading the low-grade bituminous territory during recent years. Each type has its peculiar advantages which make it acceptable under certain conditions. Generally speaking, high-ash bituminous coals handle better on chain-grates. This gives them the advantage of a large territory in the Middle West and West. Indiana coals, however, have a certain characteristic which makes them difficult to handle on chain-grate stokers, as well as underfeeds. Here the Roney stoker which has been distanced in other fields by the underfeed type still has a deservedly loyal following. The forced-draft chain grate similar to the type used for burning anthracite is also breaking into the low-grade bituminous field. The Commonwealth Edison Co. of Chicago is now experimenting with a stoker of this type, burning Illinois coals. This large concern has held to the chain grate in the face of the advances of the underfeeds.

The sub-bituminous coals are found in California, Colorado, Montana, New Mexico, Oregon, Utah, Washington and Wyoming. Lignites are also found in these states as well as in the Dakotas and Arkansas.

Up until 1911 there were comparatively few stokers in the sub-bituminous and lignite territory. The Denver Tramways, however, have been using chain grates for over 20 years and there were at that time a few installations of the standard Jones stokers. In the year 1912, the Great Western Sugar Co. installed two chain-grate stokers in its Billings, Montana, plant. It had been the general opinion up to that time that beet-sugar plants could not afford stokers because they operated about 100 days out of the year. About 40 tests were conducted on these stokers and similar boilers hand-fired, burning different grades of Wyoming lignites, and Montana coals. These tests upset the old theory and several large sugar mills installed chain grates. It was found that with a 200 ft. stack giving a draft of 0.45 to 0.50 in. at the furnace, the chain grates would burn 25 lb. of dry lignite (average moisture content 23 per cent) per sq. ft. of grate per hour, giving 25 per cent more capacity and 8 per cent greater economy than the hand fires. A few years later, the multiple-retort underfeed made its entrance into these fields and has made a good showing. It is the writer's opinion that where the load is a fairly constant one for 24 hr. per day as it is the case in a beet-sugar mill, the chain grate has the advantage. Where the load varies greatly with high peaks, as a central station, the underfeed will make a better showing. This will be

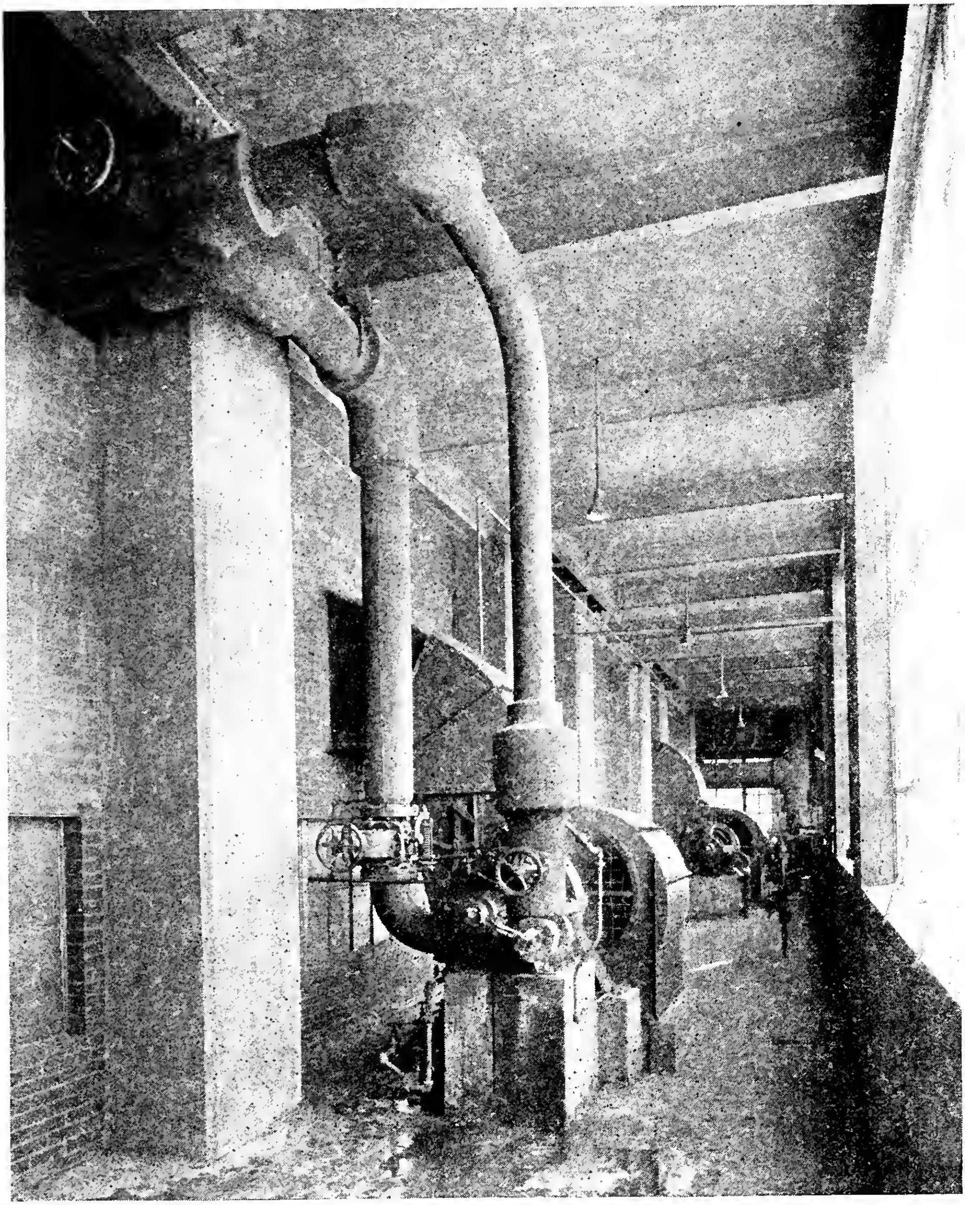

FIG. 4. ISOLATED FAN ROOM

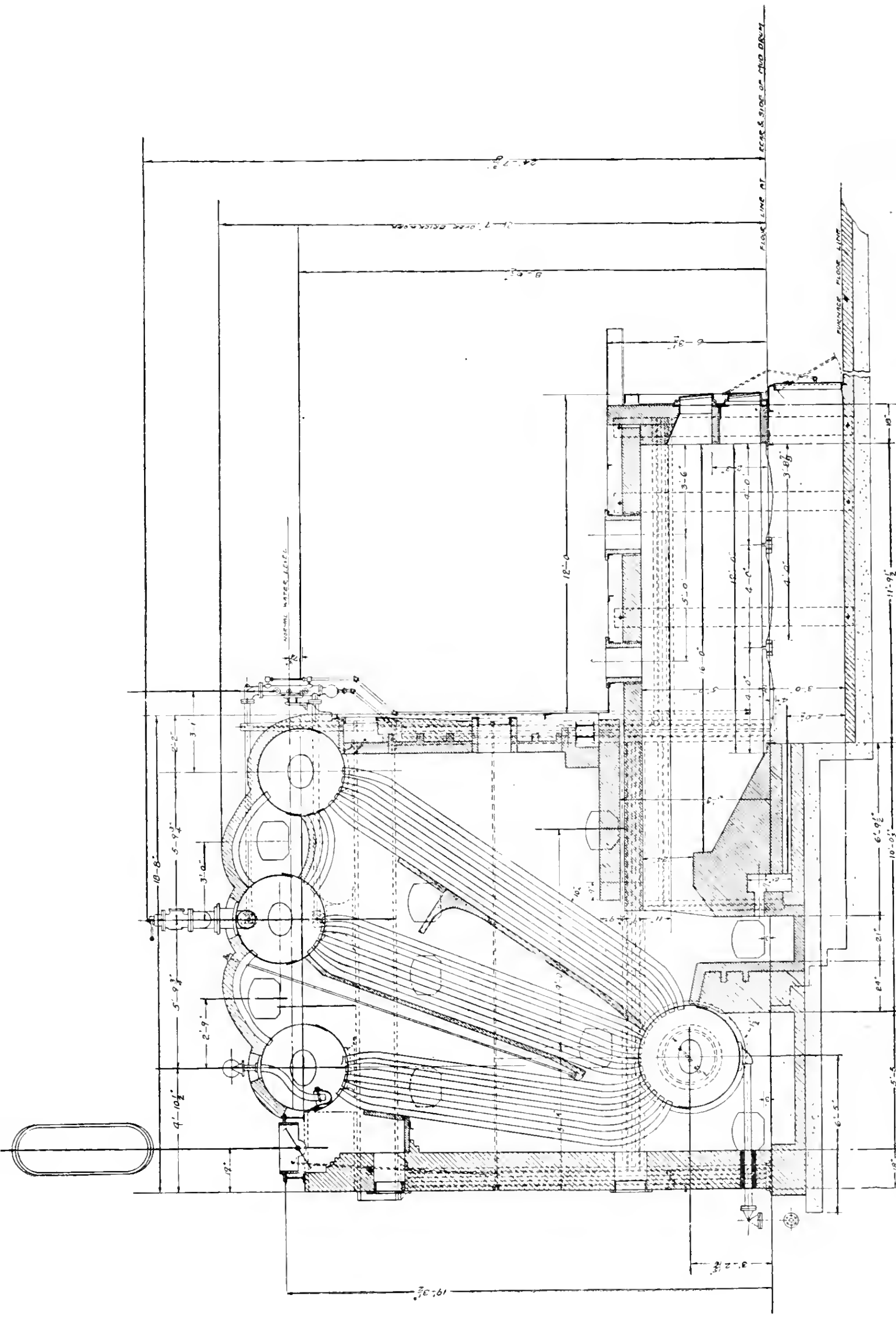

FIG. 5. SECTIONAL SIDE ELEVATION OF FURNACE FOR WET WOOD WASTE

especially true where high-ash fuels are used on the chain grate and low ash on the underfeed.

PLANT LOCATION AND FUEL SUPPLY

Item No. 2 deals with the location of the plant with respect to the fuel supply. Plants burning such fuels as blast-furnace gas or bagasse need no fuel storage. Plants removed from the source of supply should provide fuel storage of sufficient size to insure against shut down. In the oil-burning plants of the duPont Co., 40 days' storage is considered safe. This fuel comes from Mexico. In the designing of the Old Hickory plant, 25 days' supply was considered safe. This plant was less than 100 miles from the coal fields, and had priority rights over many other kinds of plants. Under present conditions, the largest amount of coal that can be stored will not be found too much.

A point which belongs under the discussion of *Character of the Fuels Used* no doubt, is the question of spontaneous combustion of coals, as it affects the method of storing coal or the character of the storage. The causes of spontaneous combustion have been discussed at length and there is still room for discussion. Generally speaking, the lower the volatile content of the coal the less the liability of spontaneous combustion. It is fortunate that some very eminent authority has said that the exception proves the rule. There are enough exceptions to the above general statement to prove it beyond any peradventure. H. H. Stoek and W. D. Langtry, prepared a short paper on Spontaneous Combustion of Coals which was distributed during the war by a department of the *National Board of Fire Underwriters.* An engineer, designing a boiler plant to burn a coal susceptible to firing would do well to follow as nearly as he can the nine conclusions of that paper, which are as follows:

1. Pile the coal to be placed in storage as shallow as possible in the available space.
2. When the coal is unloaded, if it should be lump, egg or nut, see that it is not broken by allowing it to drop a great distance and thus form smaller particles, which may be the source of trouble later on.
3. See that the coal, when it is unloaded, is piled flat—in other words, do not allow it to pyramid so that the lumps will segregate from the fine coal and thus form a chimney for the admission of air to the fine coal.
4. Pack the coal tight rather than try to ventilate it; this will prevent the easy admission of air to the inside of the pile.
5. See positively that there does not get into the pile any material such as oily waste, pieces of paper, straw, wood or anything that is of an easily combustible nature.
6. See that the coal is not wet before being placed in storage.
7. See that the coal is not piled over the top of hot pipes, manholes, or against the hot outside walls of furnaces.
8. Remember that if a fire has started in a coal pile, it is bad policy to wet the pile with water unless the fire has gone so far that it is impossible to work, due to the heat and gases being driven off.
9. If a fire has started in a pile, try to get at the source of trouble by taking the coal out and spreading it over the ground, piling it not more than 2 ft. deep, if possible; it will naturally cool down providing, of course, it is not blazing, and the coal coked until it is glowing red; if very hot it should be cooled with water.

To these suggestions may be added: where the coal is susceptible to firing, pile it first in shallow piles—say 6 ft. deep. Allow it to weather as long a time as is possible, not less than 60 days, then pile this weathered coal deeper—say 20 to 30 ft. The only way to store coal so that firing is absolutely precluded is under water. This is the only way that lignites and some sub-bituminous coals can be stored at all.

The designers of the Old Hickory power plant knew that the coal chosen for its use was susceptible to firing. For that reason, a type of storage was selected that would allow access to any part of the pile at all times. The engineers of the Link Belt Co. were responsible for this very creditable design.

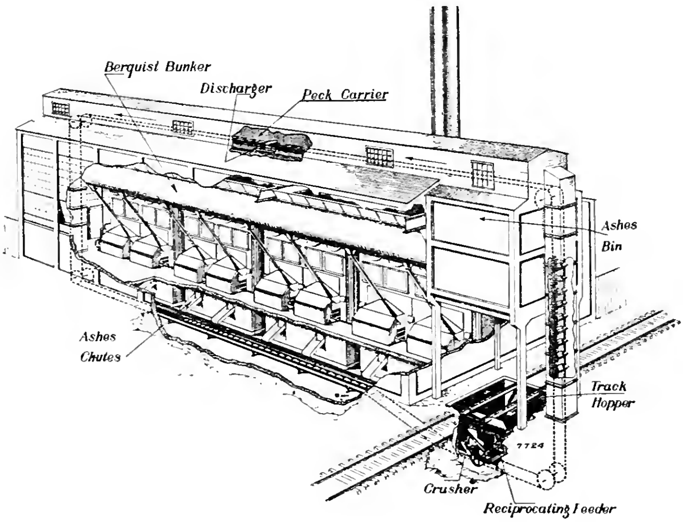

FIG. 6. ILLUSTRATING BOILER HOUSE WITH BUCKET CONVEYOR FOR HANDLING BOTH COAL AND ASHES

The third point for consideration is the character of the feedwater. The four principal things to look out for are:

1. Incrustation or scale.
2. Corrosion or pitting.
3. Foaming and priming.
4. Embrittlement of the steel.

The common scale-forming materials are calcium and magnesium carbonate and calcium and magnesium sulphate, in solution and mud in suspension. Calcium sulphate is the most offensive scale-forming ingredient. This is especially true when it is found alone or nearly so. Ohio river water is remarkably free from dissolved solids, although at certain times of the year it carries a large amount of mud in suspension. It carries about one grain per gallon of calcium sulphate. Previous to 1908, the city of Cincinnati had no filtration plant. Some boilers in the city had scale ¾ in. thick and had never lost a tube. After the filtration plant went into operation, tube failures became prevalent. Scale ⅛ in. thick was sufficient to cause blistering and ruptures. Magnesium sulphate is not an extremely objectionable scale-forming ingredient, but in the presence of calcium carbonate will exchange with the latter, forming calcium sulphate and magnesium carbonate.

In general the problem of treating water is to remove the four compounds named above. Any water containing above 20 grains of incrusting solids per gal. should be treated. There are two methods: (1) The chemical softener where the chemicals are lime and soda ash, and (2) the Zeolite method.

The following table from the L. M. Booth Co. gives the quantities required per grain per gal. for various impurities per 1000 gal. of water treated (see Table 1).

TABLE I. QUANTITIES OF LIME OR SODA REQUIRED PER 1000 GAL. OF WATER FOR REMOVAL OF VARIOUS IMPURITIES

	Lime, lb.	Soda, lb.
Calcium Carbonate	0.098	
Calcium Sulphate		0.124
Calcium Chloride		0.151
Calcium Nitrate		0.104
Magnesium Carbonate	0.234	
Magnesium Sulphate	0.079	0.141
Magnesium Chloride	0.103	0.177
Magnesium Nitrate	0.067	0.115
Ferrous Carbonate	0.169	
Ferrous Sulphate	0.070	0.110
Ferric Sulphate	0.074	0.126
Free Sulphuric Acid	0.100	0.171
Sodium Carbonate	0.073	
Free Carbon Dioxide	0.223	
Hydrogen Sulphite	0.288	
Aluminum Sulphate	0.078	0.147

Based on lime containing 90 per cent calcium oxide and soda, 58 per cent sodium oxide.

Where waters carry mud, a coagulant should be used and the water filtered or allowed to stand in a sedimentation basin until clear. Where heat is used in connection with the lime and soda treatment, the heat precipitates the carbonates, leaving the sulphates for the chemicals. The softening of water by zeolites was first introduced commercially by Dr. L. Gans of Berlin, Germany; the material marketed by him is known as *permutit.* As far as can be ascertained, it is made commercially by fusing quartz or feldspar, kaolin and sodium carbonate in an electrical furnace and then treating the mass with water. The substance resembles crushed oyster shells. Other zeolites are *decalso* offered by the American Water Softener Co.; *refinite,* a natural product in Nebraska and the Dakotas, and *barromite,* another natural product which is now being offered for sale.

Zeolite is a hydrated aluminum silicate having a formula similar to $NaAl(Si.O_2)2$. The property of softening or removing the calcium and magnesium salts from the water depends upon the replacing of the sodium atom in the zeolite by calcium or magnesium in the water. When the mineral has given up all its sodium content, and becomes saturated with calcium and magnesium, it can be regenerated by placing the mineral in contact with a 10 per cent solution of sodium chloride, whereby calcium and magnesium chlorides are formed in solution and sodium is retained by the zeolite. The cost of softening 1000 gal. of one grain water with zeolite varies from 3 to 6 cents.

Corrosion or pitting is usually caused by the presence of free oxygen, acid or sea-water. It has been found by experience that a great many waters in Colorado, New Mexico and California will cause excessive corrosion when the water is made mildly alkaline with soda ash and will stop when the concentration of soda is raised above a certain point. Commander Frank Lyon, U. S. N., finds this to be true also in marine boilers where sea water gets in through leaky condensers. A great deal of pitting in Colorado and New Mexico has been cured by heating the water to 212 deg. in properly-vented open heaters.

Priming and foaming are generally caused by sodium salts, finely divided matter in suspension, or a combination of both. The matter in suspension should be removed by filtration. The remedy for sodium salts is blowing down and should not concern the designing engineer except that where it is necessary to carry high concentrations of sodium salts, a type of boiler may be chosen which will carry the highest concentration. The Babcock & Wilcox Co. have successfully carried concentrations of 1500 to 1800 grains of sodium salts per gal. in the Stirling boiler by driving the steam from the middle to the rear drum and returning it through a set of auxiliary circulators to the middle drum.

The embrittlement of boiler steel due to the action of natural waters was first observed in a certain section of Illinois. The boilers of the State University at Urbana became affected and an investigation was made by the engineers and chemists of the faculty. The results of this investigation are set forth in Bulletin No. 94 of the University of Illinois prepared by Prof. S. W. Parr. His conclusions were that the embrittlement was caused by nascent hydrogen which is liberated by the breaking down of the sodium carbonate into sodium hydroxide and acid sodium carbonate which gives up its hydrogen atom.

He recommends as a remedy, the introduction of an oxidizing agent. If this theory be correct, it also explains why the raising of the soda concentration in certain waters that cause pitting in weak soda solutions will stop the pitting without causing embrittlement, the pitting being caused by the presence of free oxygen which takes care of the nascent hydrogen.

There has been at least one serious explosion in the U. S. Navy due to embrittlement of plates by the use of high soda concentration from compound. In treating water, therefore, it is important to remove as far as possible sodium carbonate. It is admitted that of all the boilers carrying sodium carbonate, a small per cent have suffered from the embrittling action, but the effect on these has been serious enough to warrant much thought on the subject of treating boiler water.

LOAD CHARACTER AND EQUIPMENT

The character of the load to be carried has an important bearing on the type of equipment selected for the boiler plant. Where the load runs up into high peaks for short periods of time, the underfeed stoker has considerable advantage over the natural draft types. As was mentioned above, the forced-draft chain-grate is now attracting some attention. Where the peak loads remain for a longer period of time, say for a week, as is liable to be the case, where a large heating load is carried during one of our modern winters, it is advisable to allow a more liberal sized stoker for the same heating surface. The usual custom where the multiple-retort type of underfeed stoker is used is to allow one retort

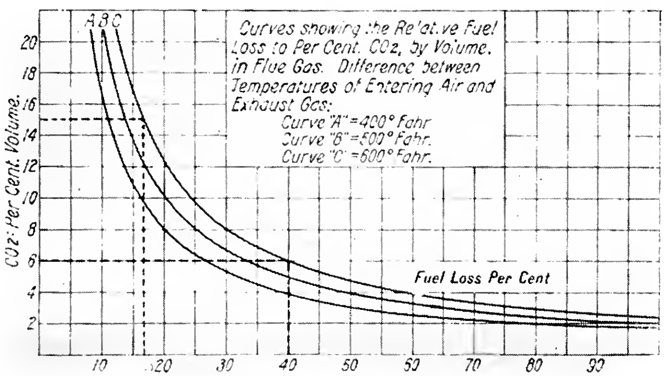

FIG. 7. CURVES SHOWING RELATIVE FUEL LOSS FOR DIFFERENT PERCENTAGES OF CO_2

per 100 rated boiler hp. Where coal is of poor quality, this ratio is reduced to one retort for 80 rated hp., and in cases where very high ratings are desired as in the Buffalo General Electric Co. plant, a ratio of one retort per 50 rated hp. or less. The usual practice in the natural-draft grates is to allow a ratio of 1 sq. ft. of grate per 40 or 50 sq. ft. of heating surface. Inclined-grate overfeed types allow a ratio of from one to 40 to one to 70, depending on the quality of the coal and on the character of the load. Chan grates cannot obtain the high capacities that the underfeeds do, but they can maintain an average output nearer to their maximum capacity than any other type within their field. Another thing which may influence the choice of a type is whether there is an abundance of exhaust steam available for heating. Where the exhaust steam cannot be used to advantage, forced-draft stokers are seriously handicapped. Comparative tests on underfeed stokers versus inclined grate overfeeds in the same plant showed that the fans consumed 4.88 per cent of the total steam generated. This, however, is an unusually high steam consumption; the figure should not exceed 3 per cent of the total steam generated. The net efficiency of the overfeeds was 65 per cent; the net efficiency of the underfeeds without credit for the exhaust steam was 67 per cent. When credit was given for the exhaust steam used in the feed-water heaters, the net efficiency of the underfeeds was 69 per cent.

If the load is carried for one shift per day, fuel economizers are hardly justifiable unless fuel is extremely high, or the owner is satisfied with a small return on the investment. Where plants are operated for two shifts per day, economizers will show a return on investment of from 10 to 30 per cent. In three-shift plants economizers will show a return of 20 to 50 per cent; this is based on today's prices in the vicinity of Philadelphia. The saving which economizers will show depends on the rate at which the boilers are driven and the efficiencies; the above savings assume ratings of 175 to 200 per cent on the boilers and good efficiencies. If the boiler efficiency is low, the economizer efficiency is high. Economizers should never be installed unless boilers are operated at 125 per cent rating or above, or unless fuel costs are abnormally high. With the high steam pressures and high superheats which are common today, economizers are almost indispensable if the load is a 24 hr. load.

If steam is to be used for generating power with turbines or engines, it will undoubtedly mean high pressure and superheat. On chemical processes as is the case with the duPont Co.'s plants, the radiation loss and expense resulting from long pipe lines for high pressure would more than offset any gain from high pressures and temperatures.

A few years ago, labor was so cheap that few industrial plants made any attempt to save labor in boiler rooms. During the war, industrial plants engaged in the manufacture of munitions expected the war to end in 6 months or less, and labor- or fuel-saving devices were not considered unless they could pay for themselves in twelve months or less.

With the outlook today, power plant designers are making every attempt to eliminate boiler-room labor. The duPont Co. operated a large boiler plant during the war (hand fired) on coal from which it obtained 300 boiler hp. per man in the boiler room. Today, it operates a plant burning fuel oil from which it obtained from 3000 to 5000 boiler hp. per man.

Mechanical soot blowers, besides saving from 1 to 5 per cent of fuel, will show a return of from 5 to 10 per cent on investment in labor saving in large plants. It requires ¾ hour for a man to blow the soot from a 600 hp. boiler with a steam lance; with mechanical soot blowers he could do a better job in 10 min. The duPont Co. learned early in the war that even though a plant operated only for one year that mechanical stokers would pay for themselves in labor and the gain in capacity.

The cost of fuel and transportation has been going up along with labor and plants, which enjoyed one dollar coal a few years ago, are now paying from $4.00 to $6.00. Naturally, the costs of boiler room apparatus has been climbing too, so that installing fuel-saving apparatus today shows little or no more return on the investment than it did six years ago. The man who put in an economical plant in 1914 or 1915, has gotten about four times the return on his investment for fuel saving refinements that he would get today. A gain of efficiency from 70 to 75 per cent in a 24 hr. 10,000 hp. plant means a fuel saving of 8250 tons annually. At $6 per ton this amounts to $49,500.

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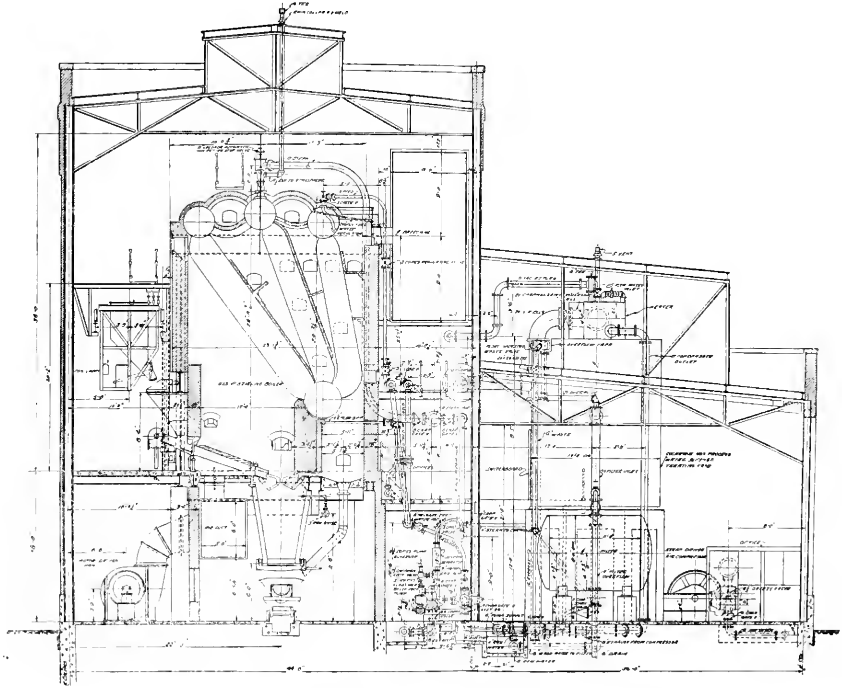

FIG. 8. TYPICAL SINGLE-AISLE BOILER PLANT WHERE SPACE IS LIMITED

It is impossible to obtain the best boiler efficiency without sufficient instruments to show what is going on. Every boiler in an up-to-date plant should be equipped with a flow meter, a draft gauge, a recording stack thermometer, and convenient means of operating the boiler damper from a position in front of the boiler where the draft gauge may be read. It is a significant fact that of all the large boiler plants in operation, the most efficient ones are equipped with CO_2 recorders or have at least some means of checking the air supply. In this connection, let us call attention

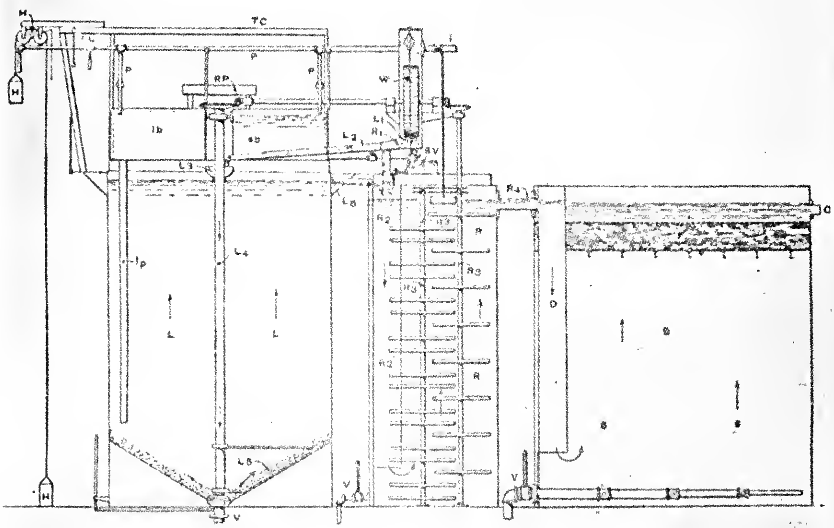

FIG. 9. CHEMICAL WATER SOFTENER; 1B and 2B CHEMICAL TANKS; INCOMING WATER DRIVING AGITATORS THROUGH WATERWHEEL W; WATER MOVING IN DIRECTION INDICATED BY ARROWS

to the Bailey boiler meter for the reason that it indicates and records all the necessary information regarding the efficient operation of a boiler.

The duPont Co. operates a boiler plant consisting of eight 823 hp. boilers, four of which are equipped with superheaters and four without. The superheater boilers have their own venturi water meter as also the saturated boilers; the fuel account is kept separate. Going back for several months the records show that superheaters actually increase the combined efficiency about 2 per cent without an appreciable lowering of the boiler capacity. Each one of these boilers burns $75,000 worth of fuel annually. A 2 per cent saving is $1500. A superheater for one of these boilers would cost approximately $4500 installed, provided it was installed along with the boiler; thus a saving of 33 1/3 per cent of the cost is realized in the boiler room.

CLIMATIC CONDITIONS AND ALTITUDE

Stacks designed to give a certain draft at sea level would not give the same draft at Denver or Cripple Creek, Colo. The barometric pressure at sea level is 1.464 times the barometric pressure at an altitude of 10,000 ft. Therefore, a stack at Cripple Creek would have to be 1.46 times as high as a stack in Philadelphia. It would also have to be 1.16 times as large in diameter. Arthur Pratt's Table 54 in the 35th edition of *Steam* gives stack corrections for altitudes from 1000 to 10,000 ft.

In very cold countries, the freezing of steam gauge and water column piping, also radiation losses and comfort of operating crew must be taken into consideration. The handling of frozen coal cars is a problem. In this connection, attention is called to the method of handling coal used by the Erie General Electric Co.

Having considered the first eight of the points on known quantities, the engineer will have to apply the other five in considerations, viz.:

9. Importance of the time element in getting the plant going.
10. Local laws.
11. Return expected on invested capital.
12. Permanence of the plant.
13. Value of the real estate to his selections, and arrangement of apparatus to see if they will stand the final test.

Having covered the thirteen qualities which must be known in a particular case, let us turn to a few things in general which apply to every boiler plant.

The laying out of the steam piping is usually not given the attention it deserves. There are two losses to be given consideration in steam piping: (1) The pressure loss which is dependent on the velocity of flow and the density of the steam. (2) Radiation loss which is dependent on the difference of temperature between the steam and outside air, the size of the pipe and the insulating value of the covering used.

The economical size of pipe therefore will be the size in which the sum of these two losses are a minimum. It is generally considered good practice to allow an average velocity of 100 ft. per sec. in main headers and proportion boiler branches to allow 60 ft. per sec. minimum velocity at normal rating with saturated steam and 75 ft. per sec. with 100 to 150 deg. superheat. Where steam turbines are used as auxiliaries, velocities of 150 ft. per sec. and with reciprocating engines half this velocity should be used. Auxiliary exhaust piping should be proportioned for 150 to 200 ft. per sec. It is best not to try to follow general rules in the layout of steam piping. A. Langstaff Johnston, Jr., has prepared an excellent set of curves which will enable an engineer to intelligently design steam piping. They are published by the Engineering Magazine Co. in connection with the hand book, *Steam Piping*.

In the past, engineers have laid too much stress on pressure drops and not enough on radiation losses. For example, a 6 in. pipe carrying 30,000 lb. of saturated steam per hr. at 200 lb. gage will have a pressure drop of 1.44 lb. per 100 ft. of length, and a radiation loss of 27,571 B.t.u.

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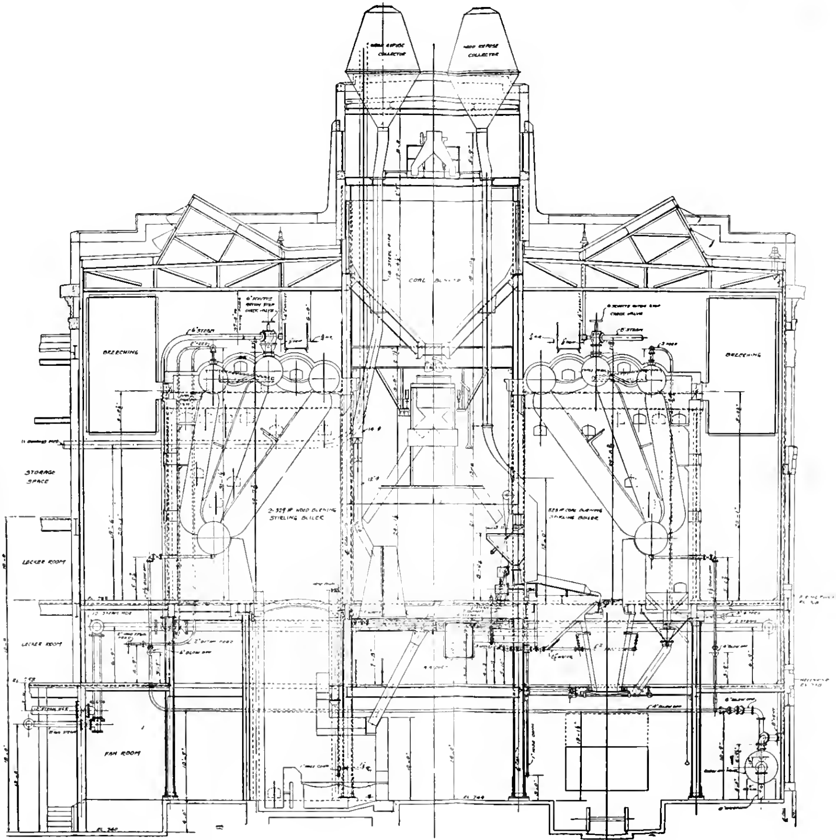

FIG. 10. TYPICAL DOUBLE-AISLE BOILER PLANT SHOWING FURNACES FOR BURNING COAL AND DRY WOOD WASTE

per hr. An 8-in. pipe carrying the same quantity at the same pressure would have a pressure drop of 0.29 lb. per 100 ft. of length and would radiate 34,531 B.t.u. per hr. Assuming that the same pressure was wanted at the end of the 100 ft. line, the boiler pressure in the case of the 6-in. pipe would have to be raised 1.15 lb. In doing this, 3300 B.t.u. would be added to the steam. Assuming for a moment that as a result of the drop in pressure, the 3300 B.t.u. were lost, the net gain by the use of the 6-in. pipe is 3760 B.t.u. As a matter of fact there is no loss of energy due to a drop in pressure, for the work of overcoming the friction is converted into heat and is taken up by the steam. On this well-known principle is based the throttling calorimeter. It should also be pointed out that the 6-in. pipe will cost 25 per cent less than the 8-in.

Where boiler units are 800 hp. or larger, they should be set single wherever coal is used as fuel. Where oil or gas is used, there is no objection to battery settings up to 1000 hp. The width of the alleys between settings will depend on the character of the fuel and size of the unit. Where oil or gas is used, 4 ft. alleys are sufficient, irrespective of the size of the boilers; where stokers with rear-end cleaning are used, alleys of 6 to 10 ft., depending on the width of the setting. The alleys should be proportioned so that a fireman can reach clear across the dump grates. Where side cleaning stokers are used, alleys of 4 ft. are sufficient provided the boilers are equipped with mechanical soot blowers.

In choosing forced-draft fans for underfeed stokers, a fan with a steep pressure characteristic should be selected. For example, when the resistance through a fuel bed is high due to ash or too-heavy fuel bed, so that the quantity of air is cut down, a fan whose static pressure will rise rapidly with a falling off in volume is the ideal fan. As to capacity, a fan should be selected for the highest efficiency under normal operating conditions based on 230 cu. ft. of air per min. per lb. of coal burned, and at the static pressure corresponding to the normal fuel-burning rate. Fans so selected will usually take care of required overloads. If such is not the case, it is better to install additional fans rather than try to choose fans to operate through such wide range of conditions. In providing fan capacity for peak loads, it is not necessary to allow 230 cu. ft. of air per min. per lb. of coal burned. This figure takes into account duct leakage and poor furnace conditions. At peaks the duct leakage is very little more than under normal conditions and the fires will have to operate at minimum excess air, so 200 cu. ft. is plenty to allow at peak load.

Generally speaking, heaters are too small. Heaters should be chosen for peak conditions. In other words, if the average load is 6000 hp. with peaks of 10,000 hp., put in a 10,000 hp. heater. The open type is usually the best for all conditions. Where open heaters are used, they should be elevated above the pumps to give sufficient head to prevent pumps from losing their suction. With 212 deg. water heaters should be 10 to 15 ft. above pump suctions. The velocity of water in suction lines should not exceed 200 ft. per min. and in the feed lines, the velocity should not exceed 500 ft. per min.

In designing stacks, the height should be determined by the intensity of draft required. Any responsible boiler builder can give the draft loss through his boiler for various quantities of gases passing through it; also the probable temperatures of the escaping gases. Assuming a maximum rating at which it is desired to operate, find the draft loss through the setting. To this loss add the loss through the flues which will be approximately 1 in. per 100 ft. of length plus 0.05 for each right angle turn. The sum will be the draft required at the entrance of the flue into the stack. With the temperature of the gases and outside air known, the height of stack may be determined to give the intensity of draft required, allowing the same friction loss as in flues. Flues should be proportioned on the basis of 35 sq. ft. per 1000 rated hp., and the cross-sectional area of a round stack may be 20 per cent less than the combined area of the flues emptying into it. Flues should be as short and free from turns as possible. Where drafts are desired requiring stacks of a greater height than 300 ft., induced draft fans are cheaper. Such will be the case where economizers are installed in connection with boilers burning coal.

Every large plant should have reliable apparatus for keeping the following records:

1. The total amount of water evaporated.
2. The temperature at which the water enters and leaves the heater.
3. The temperature of the water entering and leaving the economizers if they are used.
4. The temperature and pressure of the steam.
5. The quantity of fuel used.
6. If oil is used as fuel, the temperature of the oil entering and leaving the heaters.

No. 597

INFLUENCE OF CONTINUOUS FIRING OF HIGHLY-VOLATILE FUEL UPON BOILER ECONOMY

By A. B. Reck, Copenhagen, Denmark
Member

In ordinary practice, the users of heating boilers can hardly be expected to brush the boiler flues more frequently than every fortnight or, at the very best, every week. Frequent cleaning becomes an absolute necessity where the fuel employed has only a moderate or low content of fixed carbon and corresponding higher content of volatile matter, as, for instance, various bituminous coals, lignite briquettes, or peat.

Therefore, while it is certainly not without interest to determine the efficiency of a boiler fired for 8, 16 or 24 consecutive hours with fuel of high volatility, it is evidently of far greater interest to determine the efficiency when a boiler has been at work continuously for a week or a fortnight without cleaning.

Consequently, it is the writer's belief that it will prove of some interest to the members of the Society to be acquainted with the results of two tests with bituminous coal, one made before and another after 14 days of continuous firing with the same fuel without intervening cleaning. The cleaning doors were sealed before the first test started, and the seals were not removed until after the end of the second test. The boiler type employed was designed primarily with a view to make it suitable to a wide range of fuels, from those of about 80 per cent fixed carbon, such as anthracite and coke, to those of a much lower content of fixed carbon with an accompanying high content of volatile matter. For the tests a bituminous coal was selected with about 50 per cent fixed carbon and 30 per cent volatile matter.

Both tests had a duration of about 8 hr., during which time neither the fire was interfered with nor any additional fuel charged into the magazine. During the entire period of 14 days between the two tests, the boiler was being worked as under ordinary conditions, with a small load at night and a normal load in the day.

The test results are given in Columns *A* and *B* in Table 1, in which the items are numbered in accordance with the Report Form for Testing of Low-Pressure Heating Boilers (see Vol. 25, Transactions, 1919, p. 146). This table shows that the over-all efficiency obtained in the first Test *A* with clean boilers, namely 77.9 per cent, is lowered by only 3

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per cent after 14 days of continuous running, amounting to 74.9 per cent in the last Test *B*. Item 46 of Test *A* shows a capacity of the clean boiler of 786 sq. ft. of steam radiation, at a chimney draft of 0.157 in. water (Item 28).

This relation between efficiency, capacity and chimney draft has been shown graphically in the diagram in Fig. 1, together with similar results, obtained in a great many other tests, made with the same kind of fuel, viz., bituminous coal, and in the same clean boiler. The curve shows a

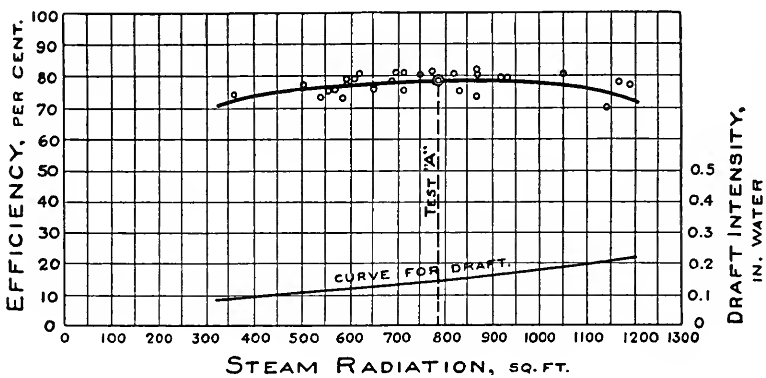

FIG. 1. EFFICIENCY CURVE FOR STEAM BOILER WITH BITUMINOUS COAL. (CLEAN BOILER)

maximum efficiency around the capacity, at which the Test *A* was conducted. As neither the capacity nor the efficiency is seriously reduced after 14 days of continuous firing at approximately the same draft (see Test *B*), the conclusion may be drawn that with sufficient data at hand of tests made with the same fuel and the same boiler after 14 days of continuous firing it should be possible to draw a new efficiency curve, in which the efficiency found in Test *B* likewise will prove to be a maximum.

A similarly small reduction in efficiency for continuous firing is recorded with other fuels of high volatility. In a test of the same boiler fired with peat for 7 days continuously without intervening cleaning, the reduction in efficiency was 3.1 per cent.

In order to verify the correctness of the high efficiency with clean boiler and bituminous coal, obtained in Test *A*, I have added to Table 1 a third Test *C*, made in the same type of boiler with the standard fuel, anthracite of stove size. The efficiency obtained in this test was 83.1 per cent, or 5.2 per cent higher than the efficiency determined in Test *A* with bituminous coal.

Such high efficiencies as are found in the type of boiler tested are undoubtedly to be attributed to the radically new method employed in introducing the supplementary secondary air. Each section of the boiler is provided with an air channel located on the edge towards the combustion chamber, as shown in Figs. 2 and 3. The air leaves the canals through two opposite rows of close sitting openings, in the form of jets, striking the combustible gases, liberated from the surface of the fuel bed, just as they are to enter the smoke flues, which are located between the boiler sections. This intermixing shower of fresh oxygen, evenly distributed over the whole surface of the fuel bed, is sufficiently far from the surface itself to avoid doing work as primary air and must necessarily result in a most intimate mixture of gases and air, immediately causing a complete combustion of the gases. This will, in turn, mean a negligible loss through the stack due to unburned gases.

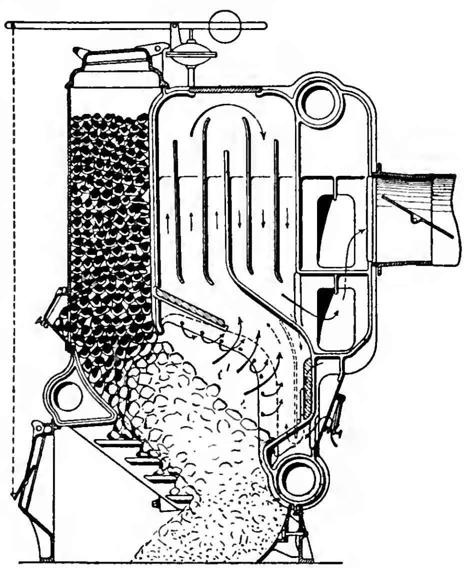

FIG. 2. VERTICAL SECTION THROUGH BOILER

Besides serving the purpose of introducing secondary air, the air channels are useful in another way. For it is generally conceded that the main cause of cracking of boiler sections is to be found in overheating of the water ways exposed to radiant heat from the fire (see for instance the paper by C. R. Honiball, Vol. 25, TRANSACTIONS, 1919, p. 397). Such overheating of the exposed water ways may be due either to deposit of scale or to a layer of steam. Now, it is evident that an air channel, constructed and located as shown in Fig. 3, must act as a protection for the exposed water ways, because the radiant heat strikes only the walls of the air channels, the heat being carried through the metal to the back of the channels and there absorbed by the water.

The description here given will be sufficient for the members to realize why it has been possible to obtain such a high efficiency with a fuel rich in volatile matter even after a long period of continuous firing.

The Tests *A, B* and *C,* given in detail in Table 1, were conducted by Professor H. Bache, Director of the Mechanical Laboratory of the Copenhagen Polytechnic Institute.

TABLE 1. RESULTS OF TESTS OF HEATING BOILERS TO DETERMINE THE EFFECT OF CONTINUOUS FIRING WITHOUT CLEANING.

	Item	Test *A*	Test *B*	Test *C*
1	Type of boiler	Reck Boiler, Type *D*	Reck Boiler, Type *D*	Reck Boiler, Type *E*
2	Grate area, sq. ft.	1.72	1.72	2.37
3	Date of test	June 3, 1919	June 17, 1919	Feb. 7, 1920
5	Duration of test, hr.	7.35	8.22	9.73
6	Times fired	1	1	1
7	Longest interval between firing	7.35	8.22	9.73

8	Fuel, name and size	English coal	Bituminous nut size	Anthracite, stove size
9a	Calorific value of fuel as fired B.t.u.	11,131	11,936	14,218
9b	of combustible, B.t.u. $= (9a) \dfrac{100}{100-(14a)-20a)}$	13,500	13,750	14,963
10	Fuel capacity, lb.	200	200	298
11	Fuel available (less re-kindling reserve)	160	160	220.0
12	Fuel available will last hr.	7.35	8.22	9.73
13	Fuel weight as fired, lb.	160	160	220.0
14a	Fuel moisture (Percentage of (13)	11.06	6.33	1.58
14b	(Weight, lb.	17.70	10.13	3.48
16	Recovered from ash-pit, lb.	3.36	1.60	8.58
17	Dry fuel burned in test, lb. = (13) — (14b) (item (16) not considered)	142.3	149.9	216.52
18	Dry fuel burned per hr., lb. = (17) ÷ (5)	19.36	18.24	22.24
19	Dry fuel burned per sq. ft. grate per hr., lb. = (18) ÷ (2)	11.25	10.61	9.38
20a	Ashes and clinker (Percentage of (13)	6.29	6.77	3.40
20b	(Weight, lb.	10.1	10.8	7.48
22	Combustible, per hr., lb. = $\dfrac{(17) - (20) - (16)}{(5)}$	17.53	16.7	20.6
23	Temperature of steam, fahr.	213	213	215
24	Temperature of feed water, fahr.	57	59	51
25	Temperature of gases, fahr.	449	576	405
26	Temperature of boiler room, fahr.	73	78	61
28	Draft intensity in smoke pipe, in. water	0.157	0.161	0.193
34	Steam pressure in boiler by gage, atmos	1.02	1.02	1.06
36	Percentage of moisture in steam	0	0	0
37	Percentage of CO_2 in escaping gases	12.9	12.8	14.1
38	Weight of water fed in boiler, lb.	1233	1275	2297
39	Weight of water evaporated, corrected for moisture in steam = (38) — (36), lb.	1233	1275	2297
40	Equivalent evaporation from and at 212 deg. fahr. = (39) × Total heat of steam above feed temp. ÷ 970	1430	1477	2685
41	Equivalent evaporation per hr. of test = (40) ÷ (5)	194.5	179.7	275.9
42	Equivalent evaporation per lb. of dry fuel burned = (41) ÷ (18)	10.1	9.9	12.4
43	Equivalent evaporation per lb. of combustible = (41) ÷ (22)	11.1	10.8	13.3
44	Capacity, total available, B.t.u. = (39) × total heat of steam above feed temp.	1,388,000	1,433,000	2,604,000
45	Capacity per hr., B.t.u. = (44) ÷ (5)	188,800	174,700	267,700
46	Capacity per hr., sq. ft. radiation (45) ÷ 240	786	726	1115
47	Efficiency of Boiler, per cent = (45) ÷ (22) × (9b)	80.0	75.9	87.0
48	Efficiency of boiler and grate, per cent = (45) ÷ (13)/(5) × (9a)	77.9	74.9	83.1
49	Losses, grate, per cent	2.1	1.0	3.9
50	Losses, stack gas, per cent	11.2	14.5	8.9
51	Losses, incomplete combustion, per cent	0.5	0.0	0.0
52	Losses, unaccounted for, per cent	8.3	9.6	4.1

As the Author found it impossible to come to the United States as he had contemplated, to present the paper, it was presented by P. H. Berggreen, who is associated with Mr. Reck in the Reck Heating Co. Mr. Berggreen, in introducing the paper, prefaced it with the following interesting remarks:

The type of boiler employed in these tests is not a laboratory product; it is not an extreme innovation. It is the outgrowth of more than 16 years' work along the boiler line in a part of the world where the fuel

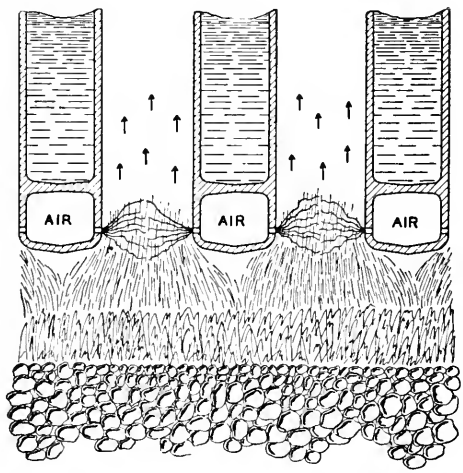

FIG. 3. DIAGRAM SHOWING EFFECT OF SECONDARY AIR UPON COMBUSTIBLE GASES

cost and the upkeep of the boiler is of far greater importance than is the case here in the United States, and where the competition among most boiler manufacturers is not keen enough to warrant the price to be the controlling factor in development. We have therefore always paid a great deal of attention to the correct design along scientific lines of heating boilers and I suppose it is only a natural event if a strictly scientific, correct boiler is originated in Scandinavia.

DISCUSSION

E. S. Berry: It was interesting to hear Mr. Berggreen say that the fuel that they commonly burn there in Denmark costs them $55 a ton, and sometimes more, depending on the rate of exchange. That casts a very interesting light upon the situation over there in the burning of fuel. And then they are limited very definitely as to the quantity each family may have, according to the number of rooms in the residence, or the space to be heated.

THE PRESIDENT: I would like to ask Mr. Berggreen if this is a smokeless boiler.

P. H. BERGGREEN: Oh, yes, of course, if it is properly taken care of.

C. F. NEWPORT: What is the purpose of the door at the bottom of the magazine?

P. H. BERGGREEN: That is primarily a firing door for starting the fire. Before the magazine is filled some fuel is piled in there on a certain amount of ashes left from the last fire, and also the fire can be observed there. We look in through here when our test is through, our fire is low and when we need more fuel.

STEWART A. JELLETT: How do you clean these flues?

P. H. BERGGREEN: We have three of those on the top, one for each section. Also on the side.

J. D. CASSELL: May I ask Mr. Berggreen how long this boiler has been in use?

P. H. BERGGREEN: This very latest type has only been in use a little over two years.

J. D. CASSELL: And you haven't burned out in that time one of the preheated air chambers?

P. H. BERGGREEN: No, not of this type.

J. D. CASSELL: I feel that this is very interesting and the man who designed the boiler surely had confidence. However, they may be using a metal that is not so susceptible to heat as most metals used in this country. It is right in the intense part of the fire—what will protect the cast iron from burning out? The other section as drawn in Fig. 3 shows the bottom of the air passage directly over the flame.

H. G. THOMAS: What is the size of the air space?

P. H. BERGGREEN: The air space is about 2½ to 3 in. wide. This was just made as a diagram. It looks more like this. (Illustrating.) You can see here that the distances are not so great as shown here. I will admit that if it was designed just like this it would be too far to go for the heat.

J. D. CASSELL: Then did it burn out designed that way?

P. H. BERGGREEN: I cannot say for sure, but I should not be surprised if there was a great distance from here to there it would burn out. But it is the way this is exactly designed, and if you gentlemen come down to the place where we have the boiler installed I would be glad to show you.

R. N. PETERMAN: May I ask what is the height of the combustion chamber from the top of the fire to the bottom of the sections on the boiler that you tested?

P. H. BERGGREEN: I do not think I can say that offhand exactly; perhaps 6 or 8 in. and it is a little more for anthracite screenings or very fine coal. Of course then you use a smaller thickness of the fuel bed.

R. N. Peterman: May I ask if in the use of the secondary air, it is for the purpose of keeping the lower sections cool or is it the idea to provide oxygen to complete the combustion of the gases after they strike the cool surface of the sections?

P. H. Berggreen: The primary object, of course, is to add sufficient oxygen to make the combustion complete.

E. H. Lockwood: Do you know whether the air which is admitted from those passages is controlled from the outside so you can regulate or diminish the amount of it? For instance, on the burning of anthracite coal I suppose it would be necessary to use less secondary air than in burning 30 per cent bituminous. Is there some control on that air entrance to the passages or is that control open the same for all of them?

P. H. Berggreen: At the back of the boiler is located the secondary air damper, which you can regulate according to the kind of fuel you are using with the rate of combustion at which you are driving the boiler.

F. D. Mensing: How would you regulate that amount of air in the furnace when you can't see the condition of the fire in the boiler?

P. H. Berggreen: You know, it is already arranged in advance. You know the kind of fuel you are using and you know that with a certain setting of the main damper, with the primary and secondary dampers fixed, you will get a certain rate of combustion.

F. D. Mensing: Is such instruction sent out with each individual boiler?

P. H. Berggreen: Yes, sir.

F. D. Mensing: Then over there you are sure of getting from the coal man the same kind of fuel continuously. We are not.

P. H. Berggreen: We buy all this coal on the calorific value basis.

F. D. Mensing: In supplying coal it must be raised 5 or 6 ft. above the floor; we have no such room in the cellar. How do you get the coal in over there?

P. H. Berggreen: I fill the pail, the largest type of pail over there, and pour into the magazine.

F. D. Mensing: Now when it comes to cleaning these tubes, how long is your cleaning rod, or whatever you use?

P. H. Berggreen: I use an ordinary metal brush with a long handle.

F. D. Mensing: And have a bend at the top?

P. H. Berggreen: No, not exactly. There must be room enough to get it in, but the handle is flexible, for that matter, so you can bend it.

C. F. Newport: May I inquire about the caking of the coal in the magazine? Isn't it liable to cake and hang up in there, or does it always feed down?

P. H. Berggreen: You can't use caking coal without proper attention.

C. F. Newport: It requires stoking there?

P. H. BERGGREEN: But we have only been interested in magazine feed boilers that have been absolutely automatic working during six, eight, ten or twelve hours, during which time we would not give them any attention.

C. F. NEWPORT: After you have been running say five or six hours you have quite an accumulation of ash down in the lower end of your grate. What is to prevent the air from passing through the ash as well as the coal?

P. H. BERGGREEN: For the simple reason that the way the air then would travel would be a longer way than the other one, and it always prefers the shortest way of least resistance. Furthermore, the ashes pack tighter than the coal itself.

J. D. CASSELL: Some eight or ten years ago, A. B. Reck, of Copenhagen, visited this Society during one of its Annual Meetings. The impressions he carried home were considered so useful, that he felt in his development of this boiler that he would like to make a contribution to the Society in his presentation here of his excellent paper by Mr. Berggreen.

Mr. Reck is quite an elderly man. He intended to come here to read his paper himself, and exhibit his boiler. His wife was stricken ill, however, and that prevented his following out this desire.

I feel under the circumstances that it is due to Mr. Reck that this Society extend to him its thanks and appreciation for the trouble he has gone to for the presentation of his paper and the exhibition of his boiler, and I would so move you, Mr. President.

The motion was seconded and carried.

A. B. RECK (written): In his discussion, Professor Lockwood intimates, that on page 243, where my paper about boiler economy terminates, there is an *obvious mistake* in Item 47 called Efficiency of Boiler Per Cent.

Nobody would be more willing than I to admit a mistake if it is real, but I think, that I have a right to demand, that the professor show the calculations, on which he bases his remarks.

In his article, Professor Lockwood has not done so, but on the contrary I will try from his own explanation of the difference between the two terms: Efficiency of Boiler and Grate, and Efficiency of Boiler, to show, that my calculation of Item 7 is confirmed by this explanation instead of being contradicted.

In explaining the difference between the two terms he states quite rightly, that the efficiency *based on combustible,* also called the Efficiency of the Boiler is a value higher than the Efficiency of Boiler and Grate and obtained by crediting the boiler with the unburned fuel found in the ash.

Now it will be seen from Items 16 and 17, Test *A* in the table, that when the 3.36 lb. of combustible recovered from the ashpit, are credited to the boiler there are burned 142.3−3.36=138.94 lb. combustible, while it is 142.3 lb. that are burned, when the 3.36 lb. combustible are not credited to the boiler. The proportion between the two different efficiencies must therefore be 142.3÷138.94, or about 1.025, but the same proportion is found to exist between the two efficiencies 80.0 and 77.9, as indicated by the Items 47 and 48 in the table.

Now Professor Lockwood does not object to an Efficiency of Boiler and Grate of 77.9 per cent, as given in Item 48. It is then difficult to see, why he can find fault with an Efficiency of Boiler of 80 per cent, as given in Item 47, although this item stands in the right proportion to Item 48. As to the two other tests, *B* and *C* calculations will give similar results

E. H. Lockwood (written): On page 248, A. B. Reck requests my calculations to prove that a mistake was made in Item 47 of his boiler tests presented on page 243. I am glad to comply with his request. It is proper to say at the outset that my reference to Mr. Reck's mistake was made as a timely example, to illustrate the point that computing errors were extremely common in Item 47. This was one of my reasons for either modifying or omitting Item 47.

It would be hardly too much to say that Item 47 may be likened to a pitfall from which only the most experienced will escape. The calorific value of the combustible should be computed from results of the chemical analysis, the calorific value of the dry fuel, and the moisture and ash from the approximate analysis. Mr. Reck has used, however, the percentage of ash from his tests, instead of the ash from the approximate analysis. The difference produced in this way may be considerable. In Mr. Reck's Test *C*, the ash for stove coal was found to be only 3.4 per cent, while the ash by analysis for this coal usually is twice this amount.

It might be pertinent, for the benefit of Mr. Reck and others interested, to present my own ideas on the *efficiency of the boiler* (or efficiency based on combustible), and its relation to the other efficiency called *over-all efficiency* (or efficiency of boiler and grate).

In computing the *efficiency of the boiler,* the attempt is made to credit the boiler with any unburned fuel discovered in the test ash. For example, if the test shows that of 100 lb. of dry coal, only 85 lb. were burned, leaving 15 lb. of ash and refuse, it is assumed that if the entire weight had been combustible, the evaporation of the boiler would have been increased in the ratio of 100 to 85, or 17.7 per cent. This increased evaporation of the boiler based on combustible must be compared with the increased calorific value of the coal, based on combustible in the coal. This last is done by the results of the chemical analysis.

Suppose the chemist reports only 10 pounds of ash in 100 lb. of coal, leaving 90 lb. of combustible. The calorific value of the combustible will be increased in the ratio of 100 to 90, or 11.1 per cent. Accordingly, the net result of the two increases is to raise the *efficiency of the boiler* in the ratio of 90. to 85. or a gain of 6 per cent. It might happen in making a boiler test, that the test ash and refuse would be exactly equal to the ash found by analysis. Our reasoning shows that, in this case, the two efficiencies would be identical. Going a step farther suppose the test ash and refuse shows *less* than the ash by analysis. This leads to the embarrassing situation of a computed efficiency of the boiler less than the efficiency of boiler and grate, which is impossible.

The efficiency of the boiler is usually computed in a way that conceals the simple relations stated above. Nevertheless, it is undoubtedly true that the efficiency of the boiler involves, in addition to the efficiency of boiler and grate, only two other quantities, the ash by analysis, and the ash and refuse by the boiler test. Consistency in the two ash determinations is no easy matter, as every experimenter knows. Since the *efficiency of the boiler* is difficult to determine accurately, and since its value is speculative rather than practical, I have advocated its omission from the boiler test code.

No. 598

THE SEMI-ANNUAL MEETING

1921

American Society of Heating and Ventilating Engineers

FOR the Semi-Annual Meeting of 1921, the Society has again visited Cleveland, Ohio, after a lapse of seven years following its former visit of 1914. Since the meeting there in 1914, Ohio Chapter has been organized and has grown to maturity; this year as host it has clearly vindicated the judgment of the Council of the Society in selecting Cleveland as the location for the Semi-Annual Meeting, by planning a meeting which has broken all previous records of Semi-Annual Meetings with regard to both professional sessions and entertainment. The Meeting was an overwhelming success from every point of view and is a remarkable tribute to the growth in strength of Ohio Chapter.

Headquarters were established at the Hotel Winton on Prospect Avenue and East 9th Street, with registration headquarters convenient to the lobby and the professional sessions were held in the Rainbow Room, which served also for the entertainment on Tuesday evening. The banquet was accommodated in the Cameo Room in the Hotel, the large ballroom on the mezzanine floor.

The professional sessions of the Meeting were characterized by the presentation of many papers of widely diversified interest and the committee reports and reports of progress from the Research Laboratory proved of unusual interest. A notable action of the Society at this Meeting was a resolution in which it went on record as favoring the treatment of all bituminous coal for heating purposes in by-product coke ovens for the purpose of not only recovering all the valuable by-products, but also of transforming the fuel into coke for its advantages as a smokeless fuel. Another important action of the Society at this Meeting was the attention given to the proposed code for testing heating systems, towards which definite progress was made. There were four professional sessions held which were devoted to the following topics: Air Conditioning; Heating; Ventilation, and Research.

One of the most enjoyable events occurred on Wednesday, the 15th, when an automobile tour to points of interest throughout the city and an inspection of the Joseph Feiss Co. plant, manufacturers of Klothcraft Clothes, where guides met the members and guests on arrival and conducted them through, explaining the various interesting operations and the marvelous system employed in operation.

The plant embraces such conveniences as a swimming pool, Turkish baths, bowling alleys, steel lockers especially heated on rainy days to dry the employees' clothing, and a splendid auditorium to which the various groups returned after inspecting the plant. Here Mr. Phegley introduced I. E. Brook of Lockwood-Greene Co., engineers for the very complete heating system with its many special features necessarily installed in the plant. Mr. Brook told of the coal handling system, ash removal methods and described the heating and ventilating apparatus, air washers and refrigeration system that had been viewed by the visitors. Following this interesting talk, Mr. Phegley expressed thanks to the Joseph Feiss Co. for their courtesy and treatment of the guests, and President Riley congratulated the local committee for arranging such a highly interesting and instructive visit to the plant.

Everyone expressed appreciation for a most enjoyable and pleasant meeting in Cleveland so that all would recall this pleasant meeting each member and guest was given a beautiful book, dedicated to the memory of John R. Allen, President of the Society in 1912 and first Director of the Research Laboratory, containing photos of the Society officials, and profusely illustrated with fine prints of Cleveland buildings and show places.

PROGRAM OF THE SEMI-ANNUAL MEETING, 1921

FIRST SESSION

Tuesday, June 14, 2 *P. M.*

BUSINESS SESSION:

Welcome Address.

Response by President.

Annual Reports of Chapters:

Illinois.	Minnesota.	Eastern Pennsylvania.
Kansas City.	New York.	Pittsburgh.
Massachusetts.	Western New York.	St. Louis.
Michigan.	Ohio.	

Report of Committee on Chapters.

Report of Committee on Code for Testing Heating Systems.

Report of Committee on Code for Testing Low-Pressure Heating Boilers

Report of Committee on Increase of Membership.

Report of Committee on Model Chimney Ordinance.

Report of Committee on Revision of Constitution.

SECOND SESSION

Wednesday, June 15, 9 *A. M.*

Air Conditioning Session:

Paper:

Resistance of Materials to the Flow of Air, by A. E. Stacey, Jr.

Paper:

Drying as an Air-Conditioning Problem, by A. W. Lissauer.

Paper:

Drying of Fruits and Vegetables, by Ray Power, U. S. Dept. of Agriculture.

THIRD SESSION

Thursday, June 16, 10 *A. M.*

Heating Session:

Paper:

By-Product Coke Ovens and Their Relation to Our Fuel Supply, by E. B. Elliott.

Paper:

Hot-Water Heating in Asia Bank, by H. L. Alt.

Paper:

Circulation Problems in Hot-Water Heating, by A. W. Luck.

Paper:

Fractional Distribution in Two-Pipe Gravity Steam-Heating Systems, by J. A. Donnelly.

Paper:

The Application of Gas to Space Heating, by Thompson King.

FOURTH SESSION

Thursday, June 16, 2 *P. M.*

Ventilation Session:

Paper:

Some Developments in Centrifugal Fan Design, by F. W. Bailey and A. A. Criqui.

Paper:

Plea for Better Distribution in Ventilation, by J. R. McColl.

Paper:

Ventilation in Metal Mines, by D. Harrington, U. S. Bureau of Mines.

FIFTH SESSION

Friday, June 17, 10 *A. M.*

Research Session:

Report of Committee on Research.

Paper:

Apparatus for Testing Insulating Materials, by F. B. Rowley.

Paper:

Ventilation Tests of School Rooms at Minneapolis, by L. A. Scipio.

Paper:

Chimneys for House-Heating Boilers, by L. A. Scipio.

Paper:

Dust Standards, by S. H. Katz.

Paper:

Commentary by L. A. Scipio, on Report on Heat Transmission, Corkboard and Air Spaces, by A. J. Wood and E. F. Grundhofer.

No. 599

DRYING OF FRUITS AND VEGETABLES

A Preliminary Study of the Effect of Different Relative Humidities on the Rate of Drying and Quality of Product

By Ray Powers,[1] Washington, D. C.

Non-Member

THE proper dehydration of fruits and vegetables presents many problems. Many different factors affect the quality, which consequently affect the value of the finished product, and the actual cost of production, both being of economic importance to the commercial drying plant.

The production of dehydrated products of good quality is largely dependent on careful attention to each step of the process—preparation, blanching and drying. The actual drying or removal of the moisture is but one of three important steps, and must be accomplished without damage to the appearance, texture, and flavor of the product. It is also desirable that the moisture be removed as rapidly as possible, since the products tend to spoil if drying is prolonged. A decrease in the length of the drying period, furthermore, increases the capacity of the plant. It is very important, therefore that we know the optimum conditions for drying each product.

In atmospheric driers there are three important variables—temperature, relative humidity and circulation. In evaporating water from a free surface, dry air circulated at a high temperature and high velocity gives the quickest results . However, the nature of the material limits the temperatures which may be applied and in the case of fruits and vegetables having a cellular structure the other conditions most efficient for the evaporation of water from a free surface do not hold.

The phenomenon of *case hardening* during drying has been observed in some products, but in others this phenomenon is not shown. The term *case hardening* implies that the outer portion of the particle or product in the drying chamber has become dry and this dry layer prevents the easy escape of moisture from the innermost cells—thus greatly prolonging the drying period. Drying in a moist atmosphere tends to prevent the hardening of the surface layers of cells and allows the drying to proceed more uniformly, but in some cases the higher relative humidity may affect the quality of the product.

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[1] Commercial Dehydration Laboratory, Bureau of Chemistry, U. S. Department of Agriculture.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, June, 1921.

EXPERIMENTAL

The experiments reported were conducted under semi-commercial conditions in an experimental drier located in a large drying plant.

Drying Equipment Used. The drier used for these experiments was of the tunnel type arranged with a swinging door permitting the direction of the heated air across the produce to be reversed whenever desired. The fan used was of the paddle wheel type, attached to a single speed motor. The velocity of the circulation, therefore, could not be varied, and was constant for all experiments. The rate of circulation was determined by repeated anemometer tests, and found to be approximately 156 linear ft. per min.

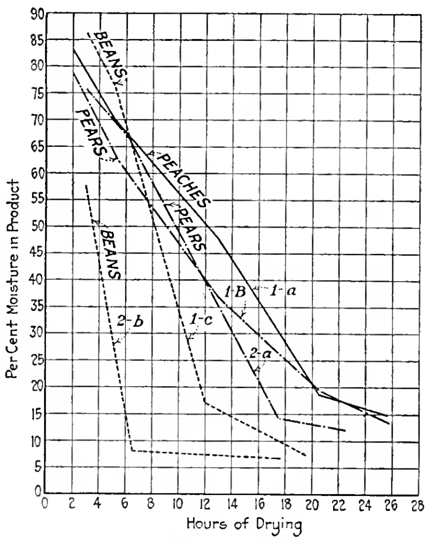

FIG. 1. SHOWING DECREASE IN MOISTURE CONTENT DURING DRYING PERIOD.

The source of heat consisted of steam-heating coils between which air from the outside was drawn. As the coils were not arranged with thermostatic control, it was necessary to increase or decrease the temperature by hand valves. In each experiment, however, when once obtained, the temperature was maintained as nearly constant as possible. This was found to vary somewhat during all drying periods, and has, accordingly, been taken into consideration in the interpretation of results. The humidity was increased by three perforated steam pipes so arranged that steam could be blown into the heated air by opening a valve outside the drier. No means was available to remove moisture from the air, but this was not necessary since the air was not recirculated.

Products Used. Peaches, pears, and stringless green beans were used in the experiments. The rate of drying under different conditions was ascertained by removing portions of material from the drier at intervals and determining the per cent of moisture in these samples. Two series

TABLE 1. TEMPERATURE AND HUMIDITY MAINTAINED DURING TESTS

Drying chamber conditions for different experiments.

Experiment	1-*a*—Peaches		1-*b*—Pears		1-*c*—Beans		2-*a*—Pears		2-*b*—Beans	
Time	Temp. deg. F.	% rel. Humid.	Temp. deg. F.	% rel. Humid.	Temp. deg. F.	% rel. Humid.	Temp. deg. F.	% rel. Humid.	Temp. deg. F.	% rel. Humid.
Start	143	18	143	18	136	18	135	86	135	86
After 1 hr.	..	..	..	..	..	..	129	79	129	79
" 3 "	133	26	133	26	133	32	138	68	138	68
" 6 "	..	..	..	..	..	..	143	63	143	63
" 8 "	135	26	135	26	135	32	141	45	141	45
" 10 "	..	..	..	..	129	21	..	..	..	..
" 12 "	134	33	..	..	..	..	..	..	..	..
" 16 "	137	21	..	..	..	..	142[2]	33[2]	142[2]	33[2]
" 20 "	134	22	134	22	132[1]	24[1]	..	..	..	..

[1] 19 hours.
[2] 17½ hours.

Note: Air flow of approximately 156 ft. per min. for all experiments.

of experiments were conducted. In the first series a comparatively low relative humidity was used, while in the second series a high relative humidity was used. It was attempted to secure approximately the same temperature in both series, thus leaving the relative humidity as the only variable.

Experiment 1-a—Peaches. The peaches for this experiment were peeled by a boiling lye solution and washed by means of cold water. Following peeling and washing, the peaches were halved and pitted by hand. They were then trayed and processed with sulphur and steam and placed in the experimental drier.

Experiment 1-b—Pears. The pears for this experiment were washed, peeled and cored by hand. Following this they were trayed after which they were sulphured and steam processed. They were dried in the experimental drier with the peaches.

Experiment 1-c—Beans. The stringless beans used for this experiment were of ordinary maturity and size. The beans were trimmed, washed, and cut transversely into small pieces. The beans were then blanched and introduced into the drier.

Experiment 2-a—Pears. The pears used in this experiment were prepared in a similar manner to those used in Experiment 1-*b*. The drying in this case, however, was conducted under very humid conditions.

Experiment 2-b—Beans. The beans used for this experiment were overgrown and rather tough in texture. They were prepared for drying in a similar manner to those used in Experiment 1-*c*. The drying in this case, however, was conducted under very humid conditions.

Results. Table 1 shows the conditions obtained in the drier in the different experiments. Table 2 shows the decrease in moisture content during the drying period, and Fig. 1 is a graphic presentation of the data of Table 2.

TABLE 2. PER CENT OF WATER IN PRODUCTS AT DIFFERENT STAGES OF DRYING

Time of Drying Hr.	Ex. 1-*a* Peaches	Ex. 1-*b* Pears	Ex. 1-*c* Beans	Ex. 2-*a* Pears	Ex. 2-*b* Beans
2	83.1	78.8	....	....	....
3	....	....	86.31	75.58	57.82
5¼	69.55	62.52	75.26	....	....
6½	....	....	....	65.72	8.00
12	....	....	17.02	....	....
13	47.90	36.55	....	....	....
17½	....	....	....	14.25	6.70
19½	....	....	7.29	....	....
20½	18.77	19.42	....	....	....
22½	....	....	....	12.08	....
25¾	14.89	13.60	....	....	....

COMMENTS

The pears dried in Experiment 2-*a* were perceptibly darkened as compared with those dried in Experiment 1-*b*. This can hardly be attributed to scorching or too high temperature since the average temperature in 2-*a* was lower than that of 1-*b*. The results with pears indicate that drying at high humidity is injurious to quality.

Unfortunately no comparison of high and low humidities was made with peaches, but the data are included because in a dry atmosphere the drying rate of peaches would appear to be approximately the same as in the case of pears. Both peaches and pears were dried in halves and the size of the pieces would therefore be approximately the same. Both products contain a considerable percentage of sugar, but are quite different in texture. The fact that the drying rate is similar, is interesting.

In the case of pears, from the data secured, the difference in relative humidity would appear to have little effect. The rate of drying does not show any advantage for either high or low relative humidity. As noted previously, however, the high humidity does appear to injure the quality of the product. Fig. 1 indicates, however, that under conditions of high humidity, the removal of water is at first retarded, but after the first five hours, drying proceeds more rapidly than under dry conditions

The beans appear to dry much more rapidly in an atmosphere of high relative humidity. The loss of moisture in this case was distinctly more rapid under conditions of high relative humidity. This suggests that in the drying of beans it will be advisable to maintain a high humidity in the drying chamber. The higher humidity in this case did not appear to have any deleterious effect on the quality. The water is removed more rapidly from beans than from pears or peaches. This may be due to the fact that beans are in a more finely divided condition.

The experiments reported are to be regarded as preliminary, and the results as only suggestive. Our knowledge of the role of humidity in the drying of fruits and vegetables is at the present time very meager and much experimenting must be done before any final conclusions can be drawn. The results reported in this paper indicate that the problem is more difficult than is usually believed, since it appears that we must determine the effect of different humidities on each product. There are many other factors which affect the drying rate in addition to relative humidity, and all conditions must be controlled if results are to be reliable.

The study of these factors under semi-commercial conditions, where close control is impractical does not appear advisable, and it is believed these studies could be made to better advantage in a specially designed experimental drier where all conditions can be closely regulated.

SUMMARY

As previously indicated the experiments reported are to be regarded as preliminary, but the following tentative conclusions appear reasonable:

1. The results indicated that pears are injured in appearance when dried in an atmosphere of high relative humidity.

2. The drying rate of peaches and pears in relatively dry atmosphere was approximately the same.

3. Different relative humidities do not appear to affect the drying rate of pears.

4. Results indicated that beans dry more rapidly in an atmosphere of high relative humidity.

DISCUSSION

J. E. Whitley (written): In California, where the conditions are especially favorable for fruit and vegetable growing, and where these conditions have placed that commonwealth at the head of the list among the fruit-preserving states, those commercially interested in the industry have formed the Pacific Dehydrators' League. There are already several organizations devoted to special branches of the trade and the league is intended to form a central body for the discussion and promotion of all problems connected with this growing industry. The officers chosen at the initial meeting are: *President,* J. B. Howell, San Francisco; *First Vice-President,* Clarence Grange, Yountville; *Second Vice-President,* J. T. Cowles, Santa Rosa; *Secretary-treasurer pro tem,* C. A. Sheppard, San Francisco.

The program of the league is a very broad one, and is formulated on a trade basis. It is not only to act in educational work among the people at large, but it is to bring about legislation intended to protect and foster the whole food industry as it is affected by dehydration. Just now, when hastily considered tariff schemes are being rushed through, it is important that the status of the dehydrated material be definitely fixed, not only from the consumers' point of view, but for the protection of the farmer and ranch owner as well.

One idea of the league is to build up a large export business, and this particularly appeals to the California producer, where the output may be increased many fold. It is possible also that other states where alert up-to-date departments of agriculture exist, may see the possibilities of the situation, and build up a profitable line of business for their own people, and, when the technique of the trade has been put on a more scientific basis an opening will be made for specially trained engineering talent.

The Hoover plan for a thoroughly exhaustive study on a world-wide basis of the entire food problem opens up a tremendous prospect of possibilities for the dehydration engineer. His several years experience in trying to feed the starving world when nation after nation was obsessed with the fighting craze and allowed its household affairs to get into chaos gave Mr. Hoover a wealth of experience available to no other living man. He had collected a mass of valuable data, and was seeking a world-wide plan, which would make such a debacle impossible in the future. He interested the Carnegie Foundation in his plans, and that concern has placed at his disposal the substantial sum of $700,000 to carry out his plans. He placed his gathered material in the custody of his alma mater, the Leland Stanford University of California, and on July 1st the project will take definite form under the control and direction of three carefully selected experts. Dr. Alsberg was taken from his position as chief chemist of the United States at Washington, and he has associated with him Dr. Davis, of Harvard, and Dr. Taylor, of the University of Pennsylvania. Each has been assigned an angle of the vast problem where he has already made a record and definite results may soon be looked for.

Stated in a few words, it may be said that the query is how and where may the world be fed. Somewhere there is enough for all. It is grown or may be, and then the job is to get it to the hungry waiting mouths. Eliminate waste, and the whole problem is met. In this country, a tremendous amount of energy is misdirected. Land and sea furnish a liberal supply of foodstuffs, but it does not reach the ultimate consumer. Ship-loads of fish are tossed overboard, and train-loads of vegetables and fruit, the result of much effort and expense, are allowed to stand and rot on the side-tracks. The facts, when told, are almost unbelievable, but they go on piling up, year after year, and the task of these three experts is to indicate a solution.

It might be a fair query, whether the possibilities of dehydration may not offer one avenue of escape from much of this criminal waste. The foodstuff, having been grown or gathered, may be saved for use by the cheap, simple process of drying. A fan and hot air may answer for most vegetable products, but with meat and fish the vacuum process is alone available, and it is the cheapest and most certain for all articles of diet.

The transportation wing of the general problem is thus lopped off almost entirely. In a sanitary way many dangerous poison problems disappear. In round figures it may be stated that 1 cent per lb. puts the article in shape for indefinite preservation, and in each case the vacuum chamber will take the article at its prime and deliver it in a similarly choice condition.

The Hoover-Carnegie Commission has not as yet formulated its complete program. The work has been merely outlined, and each expert will attack it from a special point. It has the whole world to work in. America is far behind other lands in selecting the best foods from an economic and dietetic point of view. Prejudice is the main obstacle to be overcome. The war and its commissary department did much to broaden the view of many of those who took part in it, and this Mr. Hoover proposes to standardize into a sensible system of provender for the entire civilian population, and his leading expert will be the up-to-date dehydration engineer.

At the recent gathering of the *National Oyster Trade* in New York City the main topic of interest was the gradual diminution of the supply of the raw material. Instead of the *sets* growing to maturity in the normal fashion of past years, they are failing to grow into edible oysters, and the experts do not seem able to diagnose the trouble and suggest a remedy. It was thought by some that the polution of the waters by sewage was the reason, but it was pointed out that many health boards are already protesting against oyster beds purposely started in sewage-infected areas, and where heretofore the animals seemed to fatten and prosper. Then, too it was pointed out that the Connecticut River with many factories along its banks, has been for years a favorite area for oyster beds. Another reason given was the succession of abnormally warm winters, whereas cold water is generally credited as favorable for oyster culture. The Fisheries Department of the United States Government has been appealed to, and that of the Canadian Government as well, but as one despondent wholesale dealer in the metropolis summed it up after the convention, "It begins to look like an oyster famine within the next five years unless something happens!"

Against this possibility, which applies particularly to the territory drawn upon and catered to by the association members, and which is now limited by the expensive and uncertain method of shipments under ice, there is the rising prospect of the coming into general use of the little known dehydrated oyster. The 90 per cent water content of the bivalve is eliminated by the vacuum system of drying, and then the finished product is ready for shipment at the lowest rates. It may be kept for months, and, when cooked, cannot be distinguished from the fresh ice-packed article. They are desiccated as all other food products are when in prime condition, and, of course, are edible anywhere and at any time. This system also disposes of that old impediment of the R-less months.

The application of this system will open up the waters of the world in the oyster quest, and make an international matter of it. Already it has been found that there is an abundance of this and similar food articles in foreign waters, and with cheap, foreign labor, the oyster-loving American may be compelled to use the imported article. From the queries coming and going it may be that the Asiatics will take over the whole industry, and with such an opening they may soon embrace the whole fish trade, placing it on a dehydrated money-saving basis instead of the present unsatisfactory fashion in which fish, both fresh and canned, are furnished the public.

J. D. Hoffman: I notice in Mr. Power's paper that the air velocity is very low, not to exceed 2½ ft. per sec. This is very much lower than one would expect as it is probably below that of the average furnace heating plant in a residence. It would be interesting to learn in another report from the author whether or not this low air velocity was necessitated because of the case-hardening or some other factor entering into the drying of the product.

W. L. FLEISHER: I would like to point out that since about 1916 the Department of Agriculture has been doing work on the drying of fruits and vegetables. They have spent more time than any individual concern or any individual engineer on this work, and they have given us, for all the work they have done, not one conclusive thing, as far as I can remember. I think that if any other private concern had spent the amount of money and the amount of time on this type of work that the Government has spent that we might have something of real value to the country. My conclusion is that they do not know how to go about it; they do not know what we want or what the public wants. The idea that they should make experiments of this kind without the proper methods of figuring the results or obtaining the results that are necessary to base some definite conclusion on is certainly a criticism of their methods.

The Society has presented to the various industries and schools definite problems that we wanted worked out which would be of assistance to the Research Laboratory. I certainly feel, from my knowledge of the personnel of the Department of Agriculture, particularly the Chemistry Department, that is working on this subject, that they need definite instructions on what we or the public want. It is foolish to go about making these experiments without calibrated instruments, without proper methods of conditioning air volumes and humidities and other things of that kind. I think that the Society or the Research Laboratory, for instance, should give their very valuable instructions to the Government as to what they should do to get results that would be of some value to the heating and ventilating engineer and the country at large, and that they can do the public a service by appointing a committee from the Research Laboratory or the Society as a whole to instruct the Government as to how to spend their money to get results.

JAMES H. DAVIS: Mr. President, I believe that the investigations that have been made on dehydration both by the Government and by private individuals have advanced far enough to show us that the product is all right. According to my observation from travel and from correspondence, what we now need is to educate the housewife. As long as the housewife knows little or nothing about dehydrated food we will not make much progress, and the large effort should be towards showing the housewife, the hotel man and the consumer that the product is all right.

W. L. FLEISHER: I do not agree with Mr. Davis as I do not believe that you can educate the public in the use of dehydrated foods unless you get a uniform dehydrated food; I mean a food which is uniformly good as turned out by the different people who dehydrate. The Government or the corps which was handling that matter stated that it would cost a quarter of a million dollars any way to get the proper publicity or the proper propaganda for the dehydrated food.

The Government have never been able to obtain uniformly satisfactory results in this problem because they have not concentrated on the correct method. The Department of Fisheries obtained a drier which would give results, calibrated, and equipped with sufficient instruments, but they have never had the funds to operate.

I wish to make a motion that this Society offer its cooperation, through the Director of the Research Laboratory, in determining data on the dehydration of fruits and vegetables.

The motion was seconded and carried.

R. G. Knowland (written): As a result of this drying investigation on fruits and vegetables, a conclusion has been reached to the effect that green beans dry more rapidly in an atmosphere of high relative humidity. Also, that pears do not dry well under conditions of high humidity. These statements are hardly adequate to express the conditions controlling the drying of fruit.

Such general conclusions should be interpreted in the light of the following facts: In the first place, the drying of almost any material, provided it has more than a very nominal thickness, is contingent upon two things; the *first* of these is the rate at which water is evaporated from the surface, and the *second* is the rate at which water diffuses from the interior to the surface where in its turn it may be evaporated. The problem of removing the water that is diffused on the surface of a vegetable is extremely simple, being merely one of passing over it enough properly conditioned air at a proper velocity to carry the water away, due regard being maintained that the temperature shall not injure the material under consideration. On the other hand, if the rate of diffusion through the material is so low that water cannot diffuse rapidly enough to maintain a moist exterior surface, then case hardening results and it is concluded that evaporation from the surface has been carried on at too high a rate. The remedy is, either to produce a condition of more rapid diffusion or else to decrease the surface evaporation. In the case of beans and similar vegetables, it may be taken almost for granted that the rate at which water diffuses through the fruit is the factor that makes the drying process slow. This being the case, if one desires to dry such materials with a maximum possible speed, he should secure the maximum possible rate of diffusion for the moisture.

The rate at which moisture diffuses through a material is very closely connected with the wet-bulb temperature and rises rapidly with the same. Therefore, if one wishes to dry beans or peaches or pears, he should, by experimentation, establish a drying schedule employing the highest wet-bulb temperature possible without injuring the fruit and evaporating the water from the surface only as rapidly as it can diffuse. Such a schedule at once prevents case hardening and produces the maximum rate of diffusion of moisture from the interior to the outside. Once the schedule is established for any given material, it may, of course, be safely followed as a regular method for control.

Now to go back to the article on the drying of fruits and vegetables, we are able on this basis to explain the fact that it was found that pears dry slowly at first under the influence of high humidity but that after a few hours drying proceeds more rapidly than under dry conditions. The reason is that when high humidity is used, the rate of surface evaporation under all conditions is relatively slow but that the diffusion of moisture from the interior during the whole drying period is excellent, provided the temperature is maintained high. In the course of five hours, without a fairly high humidity, one would get a certain amount of case hardening which in this case does not appear because the rate of internal diffusion has been maintained too high to permit it. Therefore, the drying proceeds more rapidly as a whole under high humidity conditions than under those of low humidity. In each case—*2b* and *2a,* the product dried under humid conditions reaches the lowest moisture content in less time than 1*b* and 1*c* which were dried under low humidity conditions.

In regard to the drying of beans under high relative humidity, it should be expected, as found in this investigation, that they will dry more rapidly when high humidity is maintained in the drying chamber. The same is true of pears. The conclusion that water is removed more rapidly from the beans than from the pears or peaches is well founded, due to the fact that the water has farther to diffuse in the case of the pears and peaches than in the case of the beans and, since diffusion is the controlling factor, the rate of drying is very much lower. In the case of such materials as fruit, the rate of diffusion is proportional to the square of the thickness.

In conducting drying experiments it must be remembered that the same drying conditions cannot be maintained for all varieties of fruits. But, with these general laws in mind, every fruit can be efficiently and intelligently tested, thereby determining the optimum drying conditions both as regards speed and quality.

To summarize: Drying is governed by two main factors—evaporation from the surface and diffusion to the surface. The former is controlled by air velocity and relative humidity. The latter is controlled by the cellular structure and the wet-bulb temperature. The extent to which either of these controls can be operated will depend upon injury to the product, which can only be determined by experimentation. It is entirely practical to obtain more rapid drying with the use of moist humid air than with dry air, due to the great increase of internal diffusion.

No. 600

DRYING AS AN AIR CONDITIONING PROBLEM

By A. W. Lissauer, New York, N. Y.

Member

THE purpose of this paper is to convey to the members some of the writer's ideas on drying, which may demonstrate that the phenomenon of drying, as it is known to everyone, can be explained very easily by the application of the laws governing air conditioning. In other words, that drying is merely air conditioning as both of these terms are known to laymen as well as to engineers.

As everyone knows, drying is the evaporation of moisture from substances. This evaporation, of course, is accomplished in various ways; the particular process with which we are concerned now, is that which covers the evaporation of moisture by atmospheric air heated to a moderate temperature. Those familiar with vacuum drying will see that the theories here advanced are as applicable to that method as to the air drying.

Unfortunately, it might be said, the art of drying is known by everyone. There is probably no manufacturer or engineer who has not at least once tried his hand at designing dryers. As a matter of fact, the drying art is almost in the class with the jack-of-all trades. Everyone knows so much about it that actually very little is known except to the initiated. It is supposed to be the simplest branch of all heating and ventilating, and this supposition has come down through the ages, the failures having been accredited consistently to circumstances which probably had no bearing.

In the working out and gradual improvement by rule of thumb, of drying systems, the first crude dry rooms which were hermetically sealed and heated with steam coils, gave way to rooms which were provided with openings for a gravity air circulation; these in turn were supplanted by forced hot-air circulation systems. As everyone knows, however, all three of these types are still made and probably as many of the prehistoric hot closets are being constructed today as were built 100 years ago.

In calculating the size of the equipment required for drying, the crudest methods have been employed. For instance, when the writer first became interested in drying and air conditioning, quite a few years ago, he tried to

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absorb knowledge from one of the most successful drying salesmen of that time. This salesman's was the sort of personality which sold an equipment and made it stay sold, whether or not guarantees were fulfilled; and 90 per cent of the time the sale was made without any definite guarantees either of capacity or quality. This man, whenever asked as to the amount of air required for drying, would return an invariable answer, irrespective of the material to be dried, the moisture which was to be removed, the residual moisture or the drying temperature. He would always say, "For this particular problem, I would figure an absorption of 1 grain per cubic foot of air, as my experience has shown this will be about right for the material in question." This man and his contemporaries, when they found that the air would not absorb that amount of moisture, would immediately try to rectify matters by boosting up the temperature in order to hasten the drying and to increase the absorption per unit of air. Invariably, the material dried, but often under those conditions, it was not the same material as that which went into the dryer. If this cure-all could not be applied, due to the nature of the material, the job was usually abandoned and one more black mark was chalked up against Mother Nature.

On the other hand, air conditioning has always rightfully been considered a science, the required results from which could not be determined by rule of thumb methods, similar to those used in drying, but whose laws had to be studied and their applications understood and formulated; otherwise it is universally recognized that failure will follow.

It is my purpose here to give evidence that the two arts, representing in the general mind, the extremes of true science on one hand and rule of thumb on the other, are one and the same thing, and that the laws of air conditioning are those of drying and that no drying problem can be solved, unless psychrometric laws are applied.

AIR CONDITIONING

Just what is meant by air conditioning? For the purposes of this discussion, one can take it to mean the treatment to which atmospheric air is subjected for the purpose of so regulating its temperature and humidity, that it will produce given variations of moisture contents in a given material. Certainly, this is the true practical conception of this science, as in practically all of its applications, the expense for operation and installation of a system is gone to for the purpose of treating some specific substance either in the process of manufacture or after it has been manufactured; only a comparatively few equipments are bought for any other purpose.

Everybody knows that practically all materials exposed to moist air will absorb moisture, and conversely, that practically all materials exposed to dry air will lose moisture. This merely means that most materials are hygroscopic by nature, and absorb or release moisture in proportion to the moisture contents of the air surrounding them. This is actually the basis of drying. When a substance contains more moisture than it should contain, it is subjected to air which will absorb the excess and bring down the final contents of the material to the required point.

It is conceded, of course, that the moisture which is present in the air is there in the form of a true gas, providing the temperature of the air is at or above the saturation point. It is also a well-known fact that air exerts a given total pressure and that this pressure is the sum of the partial pressures of the gases entering into its composition. That portion of the barometric pressure which is due to the gaseous moisture in the air is known as the vapor pressure and is a very definite and measurable quantity. This partial pressure due to a given quantity of moisture is constant, irrespective of the sensible temperature of the air, until of course, that sensible temperature falls below the point at which the air would be saturated. Thereafter, necessarily, moisture is condensed out of the air, and the remaining vapor exerts a lower pressure than that in the original mixture. In other words, the vapor pressure of the moisture in air is the vapor pressure at its dew point and so, if the dew point of the air is known, the vapor pressure of the air is known.

WET-BULB TEMPERATURE

When moving air, containing moisture, is brought into contact isothermally with water at a lower temperature than the air, the air is cooled, thereby giving up heat. This heat is used completely for the evaporation of moisture from the water, and this is absorbed by the air as a vapor. This exchange of heat continues until, eventually, the air becomes saturated at a temperature which represents the balance between the maximum moisture absorption capacity in the air and its heat-liberating capacity. Since, in this example there is no extraneous source of heat, the final temperature of the air, to which it is reduced by the giving up of heat used in the absorption of moisture, represents the total heat contents of the air, and is known as the wet-bulb temperature. This wet-bulb temperature is constant for any given total heat of the air, and it cannot be changed unless an addition or subtraction of heat is made. This is beautifully illustrated in the new spray dryers for liquids which are now being put on the market by this company. In this apparatus, the liquid is atomized in contact with a body of warm, moving air in an insulated chamber. By tests, the radiation from the chamber at a given air velocity and with internal and external temperature conditions fixed is determined. With this information, it is possible to predict within very close limits, the reduction in wet-bulb temperature of the air passing through the process. For instance, under certain conditions, the entering wet-bulb temperature would be, say 105 deg. fahr.; it is predicted that the radiation from the chamber under certain conditions, will reduce the total heat of the air, so that a reduction of 5 deg. fahr. in the wet-bulb temperature may be expected. Our actual temperature measurements will show a leaving wet-bulb temperature of 100 deg. fahr., showing that although the air had absorbed possibly 20 or 25 grains of moisture per cu. ft., measured at 70 deg. fahr., and the dry-bulb temperature had fallen possibly 150 deg. fahr. or more, the wet-bulb had remained constant throughout the process, only being reduced by an actual heat loss, entirely independent of the evaporation-heat exchange phenomenon.

Consider the evaporation of water exposed to moving air having a definite dew point and a definite dry-bulb temperature; in other words, water which is exposed to unsaturated air having a temperature higher than that of the water. As we have seen, the moisture contents of the air exerts a pressure, and, since this moisture contents is measured by the dew point, the vapor pressure is a fixed quantity, since the dew point of the air is fixed. Since the dew point of this air is lower than any temperature at which saturation could take place, in this example, the vapor pressure corresponding to the dew point is lower than any vapor pressure which would be exerted by the system, and may therefore be taken to be the point of lowest potential. Now, the water evaporates because it is heated by contact with the air; but also it is cooled due to the evaporation. In this case, that of pure water, the resulting temperature which the water assumes is that of the wet-bulb temperature of the air, as shown in the preceding example. At first, of course, only the surface assumes this temperature, but as the evaporation proceeds, the entire mass assumes that temperature, the heat being distributed by convection currents. Now the water heated to this wet-bulb temperature exerts a vapor pressure which corresponds to its temperature, and since this vapor pressure is higher than that due to the dew point of the air, the vapor flows from the water to the air. Evidently, if the conditions are kept constant, namely, dry-bulb and dew point of the air and the velocity of the air over the water surface, the rate of evaporation will remain constant, until all of the water is evaporated. This is true because no matter how much of the water is evaporated, the remainder is still water, and the temperature and pressure conditions inducing the evaporation will be the same with a lesser quantity of water as for the original quantity.

Now, let us add salt to the water, making it a dilute solution. This addition of solid matter changes the characteristics of evaporation, as we shall see. Everyone knows that in order to boil a solution, which means that it must exert a vapor pressure higher than the total air pressure or barometric pressure to which it is subjected, it is necessary to heat it to a temperature which is higher than that required to boil pure water. As the moisture contents of the solution decreases, the temperature required to produce and maintain the boiling increases, so that if it is desired that boiling continue, it is necessary to raise the temperature of the solution higher and higher as the moisture contents become lower and lower, or as the concentration of the solid matter increases.

The phenomenon of evaporation is analogous to that of boiling, and, as shown before, it will take place as long as the vapor pressure exerted by the solution is higher than the pressure against which it must operate. As in boiling, in order that the vapor pressure exerted by the solution may remain above that against which it must operate, that is against the vapor pressure due to the moisture contents of the air, it is necessary to increase the temperature of the solution, as the moisture content decreases, or as the concentration increases.

EVAPORATION BY AIR

Now, given the same air conditions as in the example covering the evaporation of pure water, that is, air having a definite dew point measuring, a definite absolute moisture contents and vapor pressure, and a definite dry-bulb temperature, brought into contact with a salt solution.

As long as the vapor pressure of the air, corresponding to its dew point, is lower than the vapor pressure exerted by the solution, evaporation will take place, since water vapor will flow from a point of higher to a point of lower potential, the same as electric current. When evaporation starts, the solution is heated by contact with the air; its vapor pressure then rises to a point considerably above the vapor pressure of the air and moisture is evaporated; immediately thereafter, the cooling effect of this evaporation, due to the absorption of heat by the vapor, reduces the temperature of the solution. This in-flow of heat and absorption of heat, come to a state of equilibrium. Since, in the evaporation of pure water, the lowest temperature which can be attained by the water is the wet-bulb temperature of the air, it follows that the surface of the solution in question and later its entire mass, assumes a temperature which is higher than the wet-bulb temperature of the air, in proportion to its concentration, but necessarily lower than the dry-bulb temperature of the air. Evaporation continues, and the solution becomes more concentrated, tending to lower the vapor pressure exerted by it. This has the effect of tending to decrease evaporation and in the same proportion to increase the temperature of the solution by contact with the air. Continuously, in this way, a new balance between the heat absorbed by evaporation and heat in-flow due to contact is accomplished, but always with an increasing solution temperature and a simultaneous decrease in vapor pressure of the solution. As soon as the vapor pressure of the solution at the dry-bulb temperature of the air (which is the temperature towards which the solution tends) is equal to the vapor pressure of the moisture in the air, evaporation ceases. In order to start evaporation up again, it is necessary either to increase the temperature of the solution by raising the dry-bulb temperature of the air, or to decrease the moisture contents of the air; in the first place, the temperature of the solution and, consequently, its vapor pressure is increased, and in the second place, the vapor pressure against which the solution vapor pressure must work is decreased. Note here how different is the action of a solution in evaporation than the action of pure water under the same conditions.

EVAPORATION OF WATER FROM SOLIDS

Analogous to the evaporation of solutions is the evaporation of water from solids, but with some important modifications which are, in fact, complications. Solids may be considered as solutions which have approached dryness; that is, solutions in which the solids are the preponderant factor and which are no longer true liquids; further, solids may be considered as solutions which rely on capillary attraction for the uniform distribution of moisture throughout the mass instead of by convection currents, as in liquid solutions. It must be borne in mind also that this action of the capillary force is much slower than that of convection currents in solutions, and, so, in the evaporation of moisture from solids, the time element required for moisture distribution and heat in-flow is an important consideration.

The discussion relating to the evaporation of moisture from solutions may be applied as a whole to the evaporation of moisture from solids, providing that whenever the temperature of the solution is mentioned, it must be taken to mean the temperature at the surface of the solid, because of the slowness of action of the moisture and heat distributing force.

It must be granted, that in the case of a wet solid exposed to unsaturated air having a higher temperature than the solid, the temperature of the surface is higher than that of the interior. Inasmuch as it is necessary, in order to induce a flow of moisture from the interior to the surface, that the vapor pressure at the surface be less than the vapor pressure in the interior, it follows that the moisture contents at the surface must be less than the moisture contents in the interior of the material. In fact, the moisture contents of the interior must be sufficiently higher than that of the surface, to compensate for the difference in temperatures which tends to produce a higher vapor pressure at the surface than in the interior. As in solutions, the evaporation continues until the vapor pressure at the surface of the material is equal to the vapor pressure of the air, corresponding to its dew point, when the temperature of the surface of the material has reached the dry-bulb temperature of the air. This is true at any given point, if time is not allowed for the action of the capillary force, which distributes moisture from the interior to the surface. As a matter of fact, evaporation will cease entirely when the vapor pressure of the *entire* material, which has been heated completely to the dry-bulb temperature, is equal to the vapor pressure exerted by the air.

After such a balance has been established, as in solutions, in order to start up the evaporation again, it is necessary either to raise the dry-bulb temperature of the air or to lower the dew point temperature of the air, the evaporation again continuing until a second state of equilibrium has been reached. Such a series of changes in air conditions to which the material is exposed will eventually reduce the moisture in the solid to the pre-determined point.

Such a consummation is theoretically correct, but if the time element covering the dissemination of moisture is neglected, and evaporation is induced at a higher rate than possible distribution against the resistance of the material to the passage of moisture through its pores, it is quickly found that the theoretical result is not accomplished.

As shown before, there must be more moisture in the interior of the material than at the surface; since the temperature of the surface is higher than that of the interior, it follows that if the dry-bulb temperature of the air is increased too rapidly, in order to dry the solid at a given drying rate, eventually the evaporation of moisture at the surface of the material will be greater than the tendency of the moisture to flow to it from the interior, due either to capillary attraction or difference in vapor pressures, or both. This has the effect of increasing the surface temperature rapidly, so that it approaches the dry-bulb temperature of the air, and in most drying systems, materials at that temperature will bake. This baking prevents the capillary action, or in fact, the passage of any moisture to the surface and all evaporation ceases. It might be stated that more ingenuity has been expended in the endeavor to overcome this surface baking or case-hardening than on any other condition in drying. The effort to reduce this danger has given rise to the design of the so-called moist air dryer which, I believe, is rapidly supplanting the old type of hot-air dryer, where air of the same condition is brought in contact with material during all stages of the drying process.

PREVENTING CASE-HARDENING

The obvious start in a drying system is the raising of the temperature of the material, both surface and interior, thereby producing in the material a vapor pressure which will induce the most rapid flow of moisture under the given conditions. To do this heating, evaporation from the surface of the material must first be prevented in order that case-hardening may not be present before the interior heating has had a fair start. This is done, of course, by bringing the material into contact with air that cannot absorb moisture; that is, saturated air having a temperature about the same as the temperature of the air which will later be used for absorbing moisture. This heating, of course, raises the vapor pressure of the material uniformly (if time enough is given) to the vapor pressure of the air. Note that until this condition is true, the material will absorb moisture from the air. Thereafter, the vapor pressure of the air is reduced, and the evaporation proceeds rapidly. This is caused by the quick evaporation of moisture from the surface, which cools it, and reduces its vapor pressure below that of the interior, which is still heated. This extra heat in the interior, and the fact that it has a greater moisture contents than the surface, makes the interior vapor pressure considerably higher than that of the surface. This difference in pressures, aided by the capillary action, produces a comparatively copious flow to the surface, where the vapor is absorbed by the air. Woe to the principle of the moist air dryer, however, if too great a difference between the dew point and dry-bulb temperatures of the air is present; a time will come in the process when the interior and surface vapor pressures are equal. Since the interior temperature eventually will drop, the surface evaporation will be more rapid than the moisture will flow to it; the surface will then rise in temperature to approach the dry-bulb temperature of the air and the danger from surface-hardening is again present. Here is the secret then, for the successful operation of such a dryer; the drying process must be retarded by sufficient moisture in the air, the absorption rate of the air being equal to the rate at which the moisture is transmitted from the interior to the surface of the solid, due to the combination of vapor pressure difference and capillary action. Too often, a dryer is designed with the best of intentions, but after the first stage, a maximum drying rate is required, and this increase is hoped for by a boost in dry-bulb temperature and a lowering of the dew point—and case-hardening results. I might say, however, that this is not *always* the case. Capillary action is always present and many materials are porous enough, so that this force can work wonders, if time enough is given. It is my guess that capillary action is the salvation of more rule of thumb dryers than good design.

INCREASING DRYING RATE OR REDUCING DRYING PERIOD

How can we increase the drying rate, or cut the drying period, and still circumvent the case-hardening demon? Again the obvious is, that the heat required to raise both the surface and interior temperatures of the solid be supplied in such a way that the condition of the air which absorbs the moisture is not affected. Such a consummation would be ideal. We would have a material which would exert a vapor pressure independent of, and always higher than the vapor pressure of the air. As shown before, the interior vapor pressure of the material under such conditions would be higher than that of the surface, in spite of the fact that its temperature is lower; this is due to the higher moisture contents. The flow of moisture, assisted by capillary action, would then be rapid; the allowable rate of evaporation from the surface would be increased due to a constantly moist surface, and the drying time decreased. Besides this, the tendency to case-hardening would be reduced to a minimum by the use of air at a lower temperature than that required in the hot air or ordinary moist air types, which depend upon the heat of the air alone for the heat of evaporation.

This company has investigated a method which we are confident at least represents a long step towards the attainment of this ideal. That is the application of the little considered or appreciated radiant heat to air drying processes.

You have all noticed on a cold, clear winter's day the difference in temperature as felt by the body, between shade and sunshine. The air itself is just as cold under one condition as under another. You have all, riding in a railroad car in the winter time, passed by the large fires of ties or lumber often built up by the rail side, and have felt the sudden gust of heat that passed through the closed window. You have all seen the luminous electric radiators and felt their warmth at a distance, even though the room in which you were, was cold and remained so. This heat in each case is the result of the arresting by a non-transmitting, non-reflecting solid body, of vibrations similar to those of light given off by the radiating body—the sun, the fire and the heated coil. These vibrations have their place on the light scale immediately below the red, and their wave period is somewhat slower than the slowest visible light ray, but considerably faster than the heat wave. These radiant energy waves or vibrations obey the same laws as light rays; they can be reflected and transmitted. The heat from an arc lamp has been reflected by a series of mirrors, from one to the other for a surprising distance, finally being focused through a lens to set fire to paper. This shows that these waves when impinging again upon a solid body are transformed into true heat waves. Having such slow vibrations, the radiant heat is partially absorbed by substances which reflect or transmit it. The United States Bureau of Standards has made a detailed and exhaustive investigation of the transmissive and reflective efficiencies of various materials and much of this material has been utilized by this company in the application of this principle to practical designs. The Research Laboratory published the results of a series of tests by Allen and Rowley for the January, 1920, Annual Meeting, on the determination of the radiant heat given off by a direct radiator. They determined the value of the constant for different surfaces in Stefan and Boltzmann's law which states that the measure of the radiant heat transmitted from one body to another, varies directly as the difference in fourth powers of the absolute temperatures of these bodies. It was also demonstrated that a pipe coil will radiate heat from a surface which is that of a box which tightly encloses the pipes, and not from the actual over-all surface of the pipes themselves.

RADIANT HEAT ACTION

Subject the material to the action of this radiant heat, while it is in the presence of partially saturated air and we have a condition where heat is added to the material, but not to the air, so that the vapor pressure of the air is not affected, and the dry-bulb temperature remains practically the same as though there were no additional heat. The radiant energy is converted into heat at the surface of the material and eventually penetrates it. A greater percentage of this heat will penetrate to the interior, because, weight for weight the avidity of the lower temperature, more highly saturated air for moisture, is less than that of the hot dry air of the ordinary dryer and consequently a smaller percentage of the heat is used up in uncontrolled evaporation. This heat inflow brings the material temperature above that which it would assume due to the air alone, but this higher temperature will not injure the material as long as hygroscopic moisture is present in the material and sufficient water vapor in the surrounding air controls the rate of evaporation from the surface. The interior vapor pressure is thus increased above that at the surface, the rate of flow of water to the surface is increased, and the total drying process rate is faster, even with lower air temperatures and higher moisture contents in the air than can be used in the ordinary moist air dryers.

Let it not be misunderstood, that the heat due to reconverted radiant energy is different from that given up by contact with heated air. The superiority of results with the radiant heat is only due to the fact that the substance is heated by an independent source to the proper temperature, while it is surrounded by air having such a low temperature and high moisture contents that case hardening is minimized by controlled evaporation while at the same time a sufficiently low vapor pressure resistance to evaporation is attained. If it were desired to obtain the same results with hot air alone, it would be necessary to raise the dry-bulb temperature of the air to such a point that the heat available for heating the material (by drop in air temperature) would be equal to that which would be due to radiant heat alone; taking into account the low specific heat of air, for any fixed quantity of air circulated, it would necessitate the raising of the air temperature far above the required material temperature which would make baking and case hardening a certainty by enormously increasing the rate of surface evaporation to dryness; or else it would be necessary to increase the quantity of air circulated, thus introducing a factor which would make the hot air dryer uneconomical by comparison.

Remember always, that it requires a certain definite amount of heat to dry a material, and this must be supplied whether directly by radiant heat, or directly by the raising of the sensible air temperature; of course, heat is heat, but there are some ways of applying it which are better than others, when quality of product and over-all speed are taken into consideration.

VENTO SECTIONS FOR RADIANT HEAT

Up to the present time, use has been made of Vento sections, wall radiators or pipe coils for supplying radiant heat for drying. Even with the comparatively low emission rate, due to the small difference in temperature between the radiating and receiving bodies, remarkable results have been obtained. It has been demonstrated that the interior temperature of the material is higher than that due to the air, and from actual test runs, the drying time has been decreased surprisingly without any injury to the material. In fruits and vegetables, for instance, drying has been done in this way so that the re-freshened product under the microscope has exactly the same appearance as the fresh; there is no change in cellular structure, there are no ruptured cells, and in fact the products show no change in color or taste. Potatoes in slices, for instance, are now being dried in commercial quantities in air at 85 deg. fahr. dry-bulb temperature and about 60 to 70 per cent relative humidity in 3½ to 4 hr. and the product is as above described. Anyone who has had drying experience, especially with vegetables, can imagine how long it would take under these air conditions to dry potatoes without radiant heat. Split salt mackerel is being dried in eight hours with air under practically the same conditions and the addition of radiant heat, and a better and more uniform product is being obtained now than ever before. These few examples illustrate the usefulness of the radiant heat principle as applied to drying.

Sometime in the future, it is our intention to present for your consideration the design and operation of an actual radiant heat dryer which means, one that uses radiant heat and low temperature moist air simultaneously.

In closing, it is our hope that we have demonstrated the truth of the statements made in the beginning of the discussion—that is *firstly,* that all evaporation is governed by the laws of moisture, temperature and pressure; *secondly,* that as our problem leaves that of the evaporation of pure water, complications are introduced which make all the more necessary a study of the natural laws; *thirdly,* that these principles and laws are applicable to drying, and the solution of any drying problem which has for its object the obtaining of the highest quality of product with the maximum efficiency, can be accomplished if guess work is eliminated and painstaking study and experimentation be substituted.

DISCUSSION

A. A. ADLER: On page 266, the author starts with the statement, "When moving air containing moisture is brought into contact isothermally with water at a lower temperature than the air, the air is cooled, thereby giving up heat." I think that it should be *adiabatically* to make it correct. Isothermally is incorrect, because change is taking place constantly adiabatically I believe should be used instead of isothermally.

A little further along the line I think the author has not made it clear as to what we mean by wet-bulb temperature. I think he has the idea but it is not stated. I will read a portion of the sentence. It says: "Since, in this example, there is no extraneous source of heat, the final temperature of the air, to which it is reduced by the giving up of heat used in the absorption of moisture, represents the total heat content of the air, and is known as the wet-bulb temperature."

There is a distinction between temperature and total heat, and I think it would be well if the author would correct that. The paper is a very desirable one and it is worth every one's reading, and I would like to see it written out in fine shape, that is why I make the suggestion.

B. S. Harrison (written): Mr. Lissauer's description of the action of a material being dried is extremely clear. The behavior of the wet-bulb temperature in a drier heated by an indirect heater,—the heat being carried in the air—is covered by W. H. Carrier's fourth psychrometric law. "The true wet-bulb temperature of the air depends entirely upon the total of the sensible heat and the latent heat in the air and is independent of their relative proportions. In other words, the wet-bulb temperature of the air is constant providing the total heat of the air is constant."

Consequently, as stated by Mr. Carrier, the wet-bulb temperature does not change through a drier, unless the heat is lost from the drier and thereby an external cooling takes place, or extraneous heat be added. In the case of a drier having direct radiation within it, there is an extraneous supply of heat and therefore the wet-bulb temperature will not necessarily remain constant, unless this heat supply is just balanced by other external losses.

Mr. Lissauer says "subject the material to the action of radiant heat while it is in the presence of partially saturated air and we have the condition where heat is added to the material, but not to the air, so that the vapor pressure of the air is not affected."

Of the total heat given off by a so-called direct radiator, at the temperatures under discussion, only about 35 per cent is radiant heat, and the balance (65 per cent) is given off by convection. Assume, however, that the heating surface can be arranged so that it heats entirely by radiation. The heat radiates equally in all directions and strikes not only the material being dried, but the trays, racks, walls, ceiling, floor, etc. Some of the heat is reflected by these surfaces but just as soon as their temperature rises above that of the air some of the heat is given up to the air by convection so that the air temperature does not remain constant. At the most probably not over 60 per cent of the radiant heat will be given up to the drying material; the balance will go to the racks, walls, air, etc.

While radiant heat is given off in all directions its rays travel in straight lines and the effect is inversely proportional to the distance squared. Without convection, this heat cannot be controlled or distributed unless the radiating surface be so split up and distributed that it is on both sides of and equidistant from each particle being dried. Very few materials to be dried are of a nature which will allow of such arrangement in a drier, without an excessive amount of direct radiation surface and a correspondingly small free charging area in the drier.

Without convection, there is the shadow effect of the radiant heat whereby that surface farthest from the radiator receives no heat or if it gets reflected heat, it is diminished by the square of the distance ratios. Try drying a board by radiant heat. Like the heliotropic action of a flower always turning its face to the sun, the board turns its edges to the source of heat. Reverse the board and it immediately begins to warp the other way.

I have never seen a direct radiation or a combination drier which was operating successfully and which did not owe its success almost entirely to convection. I have seen a number of direct radiation driers changed from failure to success by rearranging the radiating surface so that they operated entirely by convection. With indirect heating you can, by regulating the volume of air handled and the wet and dry-bulb temperatures, control to a nicety the rate of drying.

Condensation is a heat liberating process and the latent heat given up by saturated air cooling and condensing its moisture vapor on the cooler surface of the material is far greater than the amount of radiant heat that could be safely used.

In drying any material, there are three governing conditions which must be considered:

1. The rate at which the center of the material can be supplied with the required amount of heat. This is governed by the conductance and conductivity of the partially saturated material, and the temperature which it will stand without physical or chemical changes.

2. The rate at which the moisture at the center of the material may be driven or drawn out and this depends upon another set of physiochemical characteristics of the material in question.

3. The rate of surface drying which will conform to the above and thereture which it will stand without physical or chemical changes.

Since the rate of surface drying in a dry room is a function of the dew point vapor pressure depression below the wet-bulb vapor pressure, and of the velocity of air current over the surface, with an indirect system of heating, where these factors can be accurately controlled, the system may be laid out in exact conformity to the three above mentioned governing conditions.

Vapor pressure is a function of temperature. Mr. Lissauer says: "It must be granted that in the case of a wet solid exposed to unsaturated air having a higher temperature than the solid, the temperature of the surface is higher than that of the interior." It is contrary to the laws of heat flow that the vapor pressure or temperature can be raised at the center of a material without raising the surface vapor pressure or temperature to a higher point, whether it be in unsaturated air or in partially saturated, or saturated air. You cannot pass heat through the surface to the center and at the same time, keep the surface cooler than the center. unless the surface is diathermous, which it never is.

From the point of view of the efficiency of heating surface, the radiant heat capacity of a square foot of surface in the drier would be from 1/5 to 1/15 of what its heating capacity would be as an indirect radiator under fan operation.

Let us examine a case. Take for instance one square foot of material to be dried, the material, or the wet-bulb temperature to be 86 deg. fahr. Also assume 1 sq. ft. of *radiating* surface in the drier at 248 deg. fahr. The radiant heat given off from the radiating or envelope surface would be 180 B.t.u. per hr. or 3 B.t.u. per min. Say 60 per cent of this heat is absorbed by the material in question and given to evaporation, or 1.8 B.t.u. per min. This is equivalent to 12.6 grain of vapor per min. condensed or 3 cu. ft. air per min. saturated, cooled from 96 deg. fahr. to 86 deg. fahr. With steam at 248 deg. fahr., and air at 1000 ft. velocity per min. through the heater, the 3 cu. ft. of air per min. would be heated back from 86 deg. fahr. to 96 deg. fahr., and resaturated with 0.12 sq. ft. of heating surface.

In other words, the indirect heater would be only 12 per cent of the direct radiating envelope surface in the drier or 6 per cent of the rated direct radiation heating surface required in the drier.

An excessive amount of direct radiation surface reduces the charging free area in a drier so that the output per cu. ft. of drier gross area is less than with the indirect heated drier. It is easier to provide and distribute 3 cu. ft. of air per min. per sq. ft. of material or tray surface than to provide for 2 sq. ft. of direct radiation surface.

The premises may be altered but the relations will still hold.

The claims for the efficacy of radiant heat are not new. A few years ago when direct fired, naked flame gas ovens were used for baking japans at high temperature, with indifferent results, the electric direct radiation oven bobbed up with the same claims that Mr. Lissauer makes, that is, that the radiant heat penetrates the coating to the back of it without case-hardening the surface. As a matter of fact the work is done better, faster, more economically and with infinitely better control of temperature by indirect heaters and heated air. Micro-photographs show that the surface is pitted by the radiant heat and not by the indirect heat while the latter gives better finish and lustre.

W. L. Fleisher: I might make a general statement on the subject. In the radiant heat drying that we have done, we have never applied the radiant heat without aiding it with heated air partially saturated; that is, we have never used the radiant heat alone without using convected heat at the same time.

There is not a great amount of information available on radiant heat, and none of us knows very much about it. We are now constructing practical laboratory driers, and in them we intend to try the effect of radiant heat of different intensities, to demonstrate the extent to which the drying is affected; that is, we will collect practical drying data to substantiate the claims in Mr. Lissauer's paper. We anticipate that the results will go further towards proving the efficacy of radiant heat than a discussion of a theory which is based on incomplete premises.

A good many of the statements Mr. Harrison has made, are not in any way different from our observations and conclusions, and to a great extent consist of rewording the statements made in Mr. Lissauer's paper, according to my understanding of Mr. Harrison's remarks.

As we have said, there seems to be great value in driers of this type, and though we, in common with other engineers, really do not know a tremendous amount about radiant heat yet, I can say from practical experience that within the last few months, experiments in drying fresh fish, fruits, vegetables and meats with heated air having a high relative humidity, with and without radiant heat, have shown that we get distinctly faster and better results with the radiant heat combination, than we did with the moist air alone.

No. 601

RESISTANCE OF SEVERAL MATERIALS TO THE FLOW OF AIR

By A. E. Stacey, Jr., Newark, N. J.
Member

In the processing of certain materials, it is essential that dust particles of every description be removed from the air. It is a well-known fact that spray-type air washers will eliminate only about 50 per cent of the carbon particles from the air, which necessitates the employment of some other method of dust removal.

One concern has been using a composite dry filter for the removal of carbon particles from the air. In connection with this, they wished to install an air-conditioning apparatus. In order to determine the total resistance of the completed system, it was necessary to know the resistance to the flow of air through the dry filter.

These tests were made at the laboratory of the Carrier Engineering Corp., Newark, N. J. As the results proved rather unusual and interesting, experiments were made with other materials which might be used for filters. It is the result of these experiments which are herein discussed.

With the apparatus available, it was predicted that the greatest error would tend to occur at the lower velocities through the material being tested. Inasmuch as the flow of air through orifices had been determined with great accuracy this method of air measurement was used for the lower velocities through the material. For the higher velocities through the material, the air was measured through nozzles, the velocity being determined with a pitot tube.

Resistances corresponding to velocities up to 50 ft. per min. through the filters were determined.

The assembled apparatus is shown in Fig. 1 and consists of the following: A is a 2 E Buffalo exhauster directly connected to a variable speed motor. A dry filter B was connected to the inlet of the fan, for the purpose of removing any dust particles which might otherwise affect the resistance of the filters. Between the filter and fan inlet an adjustable blast gate C was installed by which the air volume could be regulated. This was used in conjunction with the variable speeds of the motor.

The fan was connected to the resistance box by a straight 3 in. pipe and the material to be tested was mounted in the box as shown in Fig. 2. Wing nuts were used to facilitate changing the test pieces and the joint

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made tight by rubber gaskets each side of the test piece. The dimension at this point was 9 in. x 8 in. making a filter area F of 0.5 sq. ft. Static pressure tubes G were soldered in the box each side of the filter area.

The discharge end of the box was designed for mounting either a nozzle or an orifice. The nozzle made a tight slip joint with the discharge outlet and was held in place by bolts. The orifices were also bolted to the discharge outlet using a sponge rubber gasket to seal the joint.

The main nozzle H had an inlet diameter of 3-23/32 in. and tapered in the length, 9-5/32 in., to 31/32 in. outlet diameter which made an angle of convergence of 17 deg. A machined ring of proper taper was sweated on

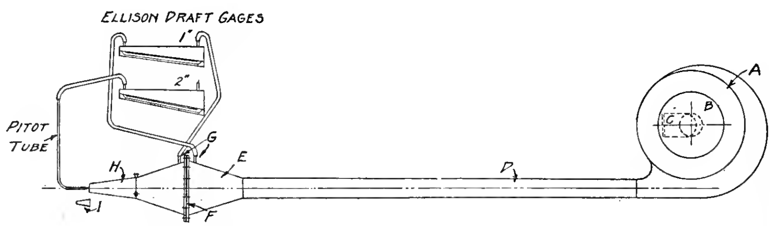

FIG. 1. ASSEMBLED APPARATUS FOR TESTING FILTERS

the nozzle for the outlet. An auxiliary nozzle I reducing the discharge diameter to ⅝ in., could be fitted to the end of the main nozzle.

NOZZLE CALIBRATION

In order to determine the coefficient of discharge for these nozzles, sharp-edged orifices were inserted at F, Fig. 1. For the 31/32 in. discharge diameter, a 13/16 in. orifice was used and for the ⅝ in. discharge diameter a ⅞ in. orifice. A special pitot tube mounted as shown in Fig. 1 was used to measure the velocity pressure at the point of discharge and the corresponding drop in static pressure across the orifice was ascertained by a differential draft gauge connected to the static tubes in the resistance chamber.

A coefficient of discharge of 0.6 (R. J. Durley, *A. S. M. E. Transactions,* Vol. 27, p. 193) was used for a sharp-edged orifice in calculating the results. Corresponding readings were taken of the velocity pressure at the nozzle and the static drop through the orifice. These results are plotted on Chart 1 and were used as a basis of calculation for air velocities through the filters in all the B tests with nozzles.

The cu. ft. per min. as calculated from the orifice test was assumed as the true value, corresponding to the velocity pressure recorded from the pitot tube and the zero point of the chart is corrected to correspond to the pitot tube readings and therefore is not the true zero. The coefficient of discharge for the nozzle can be easily calculated from these charts and is found to be 0.985. The velocity pressure readings at low points can also be corrected from this chart. These are shown to fall off very rapidly below 0.3 in. velocity pressure for both the ⅝ in. nozzle and the 31/32 in. nozzle. It is interesting to note that this point corresponds to the ratio of the areas of the two nozzles.

Several types of filters were tested which may be designated as follows:

1. Felts—Wool and cotton.
2. Cloth—China silk and cotton flannel.
3. Composite Filters—China silk and absorbent cotton; cotton flannel and absorbent cotton; muslin and absorbent cotton.
4. Metal screens.

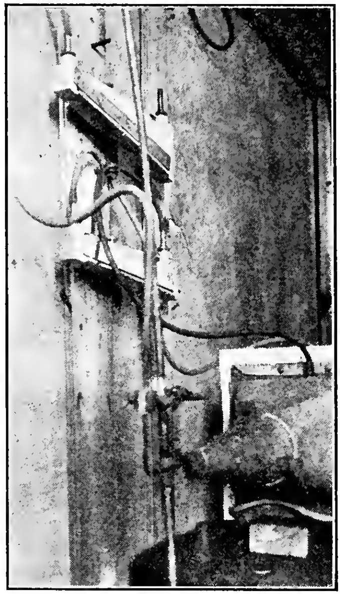

FIG. 2. VIEW SHOWING ARRANGEMENT OF PITOT TUBE AND MANOMETERS FILTER MATERIAL

In the manufacture of felts there is a wide difference in density due to the variation in the physical properties of wool and of cotton. Felt is commercially designated by thickness but it was found to be much better and more logical for purposes of experimentation to use the weight per sq. ft., as the commercial ratings were not an index to the density.

Cotton alone will not "felt" so those pieces referred to as cotton are a mixture of 50 per cent cotton and 50 per cent wool.

Experiments were made with the following felts. Two sets of felt filters were purchased at different times and are designated as A and B.

	Rating		Commercial weight per sq. ft.	
	A	B	A	B
Cotton	1/8 in.	1/8 in.	0.0595 lb.	0.853 lb.
	3/16 in.	3/16 in.	0.125 lb.	0.1255 lb.
	5/16 in.	5/16 in.	0.197 lb.	0.2265 lb.
Wool	1/16 in.	1/16 in.	0.043 lb.	0.057 lb.
	1/8 in.	1/8 in.	0.087 lb.	0.146 lb.
	3/16 in.	3/16 in.	0.127 lb.	0.156 lb.

The cotton flannel was material purchased at a small store and had 40 threads per in. The china silk had 120 threads per in. in the woof and 110 threads per in. in the warp. The finish was smooth and the threads not twisted. These two materials furnished examples of filters with a heavy nap and with a smooth surface.

Composite Filters.—The composite filters were built up of cotton fiber retained by different materials.

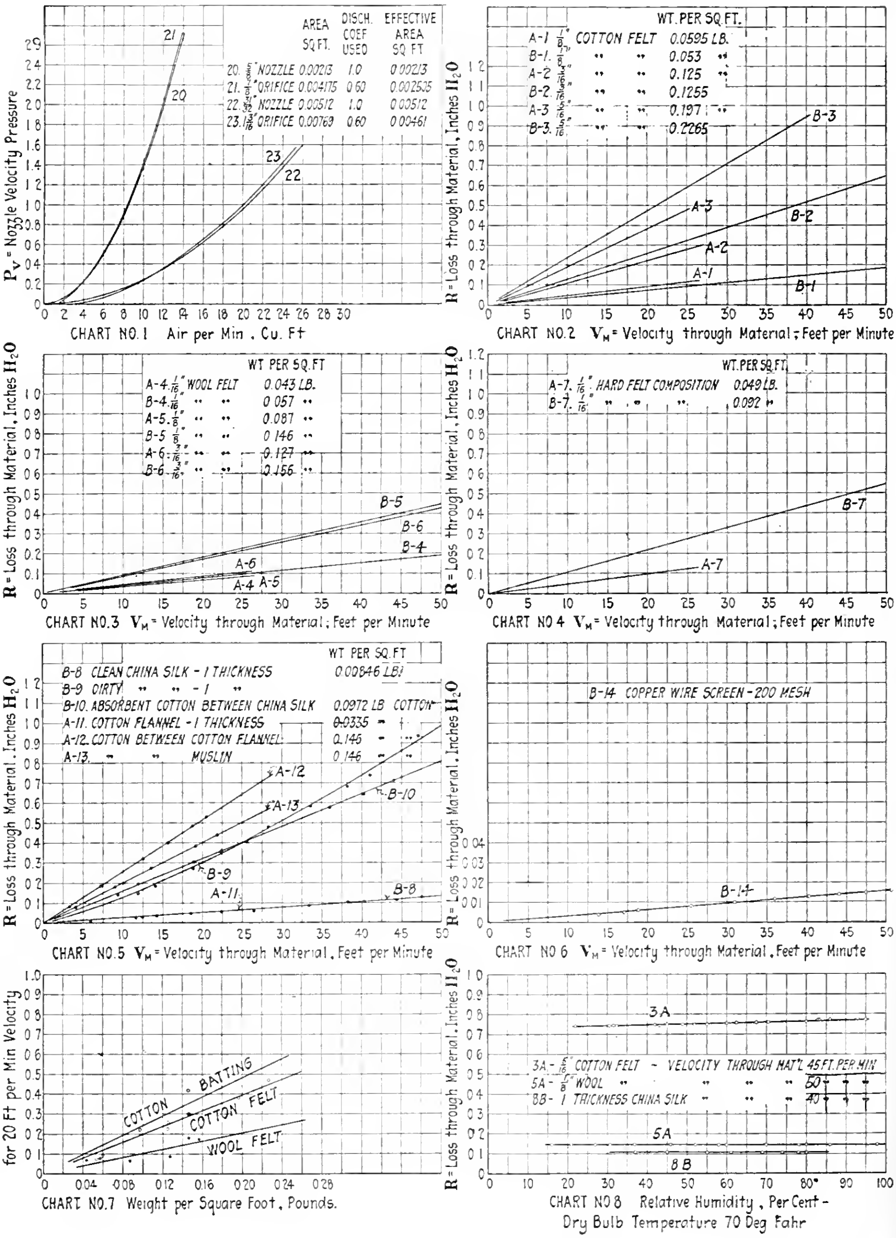

Chart 1. Calibration of nozzles with orifices.

Chart 2. Resistance for cotton felts.

Chart 3. Resistance for wool felts.

Chart 4. Resistance for hard felts.

Chart 5. Resistance for composite filters, china silk clean and clogged with carbon particles and cotton flannel.

Chart 6. Resistance for copper wire screen.

Chart 7. Relations between weight of materials and resistance.

Chart 8. Relation between humidity and resistance.

Metal Screens.—A 200 mesh copper wire screen was tested. This would be an extreme sample of a cloth filter having a smooth hard finish.

Resistance Charts.—As it is common engineering practice to design filter areas on a velocity basis, the data from these tests have been plotted with the velocity in ft. per min. through the filter for one ordinate and the corresponding static pressure loss in in. of water for the other.

As the data obtained with small orifices for low velocities proved conclusively that the resistance curves passed through zero, which is rational, and as all the curves from the test A intercepted the velocity axis at one point and those from the B test at another, a small velocity correction was made in each case to bring the curves through the zero point.

FIG. 3. VIEW SHOWING MANNER OF INSERTING FILTERS IN TEST APPARATUS

The results of the test on the china silk clogged with carbon particles were included to show the effect of service on a cloth having a smooth surface. The resistance at corresponding velocities was seven times that of a clean piece of the same material. The carbon particles filled in the interstices of the cloth and showed no tendency to cling to the surface of the threads.

On materials having a rough surface such as cotton flannel, the dust particles are caught in the nap leaving the interstices between the threads open so that the resistance of the material is not changed.

CONCLUSIONS

The curves tend to show:

1. That the resistance of air filters varies directly with the velocity of the air through the material.
2. That the resistance is not affected by variations in relative humidity.
3. That the resistance of cotton felt is approximately 100 per cent greater than wool felt of the same weight.

Charts 2 to 6 show a lineal relationship between the resistance and velocity in ft. per min. This is no doubt, due to the velocity of the air being below the critical point where it has a straight smooth flow without eddying.

These tests were all run in the conditioning room of the laboratory. This room is under both temperature and humidity control which made it possible to obtain the data on the effect of various relative humidities on the resistance there felt. The results are plotted on Chart 8 which shows that the relative humidity has no appreciable effect on the resistance

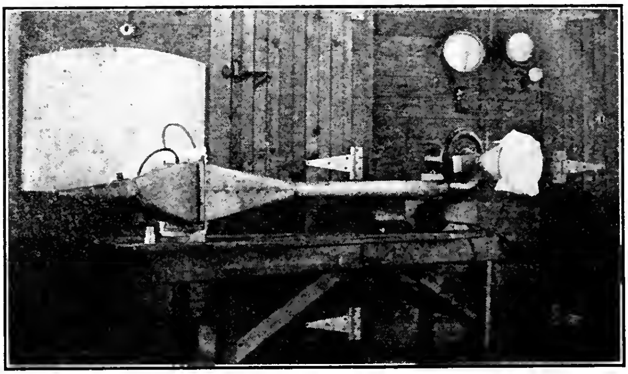

FIG. 4. GENERAL VIEW OF ASSEMBLED APPARATUS IN TEST ROOM

of the material. It is of interest to note the wide variation in the moisture content of the materials corresponding to the different relative humidities during the test. The moisture content varied:

Cotton felt .... from 3.7 to 11 per cent.
Wool felt .... from 6 to 26 per cent.
Silk cloth .... from 7.65 to 14.5 per cent.

TABLE 1. RESULTS OF TESTS WITH SILK FILTERS

Date	D.B.	W.B.	Per cent R.H.	Vel. pres. at 31/32 noz.	S.P. Loss	Vel. through Mater.	S.P. loss for 40′ min. vel.	Moist. regain %
3-1-21	70	55	3.7	1.0	0.107	40.4	0.106	7.65
	71.5	58.5	45	1.0	0.107	40.4	0.106	....
	72	61.5	55	1.0	0.107	40.4	0.106	....
	71.5	62.5	61	1.0	0.107	40.4	0.106	....
	71.5	64.5	69	1.0	0.108	40.4	0.107	....
	72	66.2	74	1.0	0.109	40.4	0.108	....
	72	67.5	80	1.0	0.11	40.4	0.109	....
	72	67	78	1.0	0.11	40.4	0.109	13.75

TABLE 2. EFFECT OF HUMIDITY ON RESISTANCE OF FELT AIR FILTER

Date	D.B.	W.B.	Per cent R.H.	Pitot tube Rdg. 31/32 noz.	S.P. loss through Material	Vel. through Material	S.P. loss for 50 FPM vel.	Moisture regain % Dry Wt.
				1/8 in. Wool Felt A-5.				
1-20-21	70.5	50.5	21	1.78	0.155	53.4	0.145	5.12
	70	56	40.2	1.79	0.155	53.2	0.145	....
	70	59	52	1.835	0.157	54.2	0.145	....
	70	55	37	1.88	0.16	54.8	0.146	....
	70	60.5	58	1.9	0.16	55.2	0.145	....
1-24-21	70.5	52.5	27	1.8	0.155	53.7	0.144	....
	70	57.5	46	1.72	0.155	52.4	0.148	....
	71	62	60	1.5	0.14	49	0.143	....
	71	64	70	1.43	0.135	47.8	0.141	....
	71	66	78	1.43	0.135	47.8	0.141	....
1-25-21	70	48.5	16	1.8	0.155	53.7	0.144	6.00
	70	56	40	1.78	0.155	53.4	0.145	....
	70	59	52	1.73	0.153	52.6	0.145	....
	70	59.5	54	1.63	0.145	51.1	0.142	....
	69.5	65	80	1.91	0.16	55.3	0.145	....
	70.7	67.5	86	1.88	0.16	54.9	0.146	11.8
	70	69.5	98	1.73	0.157	52.6	0.149	....
	71.5	71	98	1.7	0.157	52.2	0.15	26 appx
				5/16 in. Cotton Felt A-3			S.P. loss for 45 FPM vel.	
1-28-21	70	70	100	1.265	0.816	45.3	0.81	13.80
	70	68.7	93	1.41	0.826	47.7	0.78	....
	70	66.3	82.8	1.305	0.791	46	0.775	....
	70	63.5	70.5	1.30	0.776	45.9	0.76	....
	70	62	65	1.285	0.766	45.6	0.758	....
	70	61	60	1.27	0.766	45.4	0.76	....
	70.3	60	55	1.18	0.731	43.8	0.752	....
	70.5	58.5	49	1.185	0.729	43.9	0.748	....
	70.8	58	45	1.195	0.731	44	0.75	....
	70	56.5	42	1.165	0.721	43.5	0.748	....
	70.7	53.7	30	1.185	0.726	43.9	0.745	....
	70	52.5	29	1.205	0.728	44.2	0.745	....
	70	52	23	1.185	0.721	43.9	0.74	....
	70	50.8	22	1.145	0.71	43.1	0.742	....
1-31-21	71	68	86	1.085	0.72	41.8	0.774	....
	70	65.5	80	1.165	0.74	43.5	0.765	....
	71.5	70.5	95	1.2	0.76	44.1	0.775	....
	70.3	69	94	1.235	0.77	44.7	0.775	....
	70	63.5	70	1.205	0.734	44.2	0.75	....
	70	61	60	1.2	0.74	44.1	0.755	....
	70	59.8	55	1.165	0.721	43.5	0.75	....
	70	59.8	55	1.095	0.7	42.2	0.747	....
	70	59	52	1.185	0.73	43.8	0.75	....
	70	56.5	43	1.17	0.72	43.6	0.743	5.95

TABLE 3. RESISTANCE IN IN. OF WATER OF AIR FLOW THROUGH FELT—A TESTS

Vel. press. through ⅝ noz.	Vel. through Material	⅛" Cotton 0.0595 lb. per sq. ft.	3/16" Cotton 0.125 lb. per sq. ft.	5/16" Cotton 0.197 lb. per sq. ft.	1/16" Cotton 0.043 lb. per sq. ft.	⅛" Cotton 0.087 lb. per sq. ft.	3/16" Cotton 0.127 lb. per sq. ft.	1/16" Hard felt 0.049 lb. per sq. ft.
0.08	5.14	0.011"	......	......	0.005"	......	......	......
0.09	5.4	......	......	......	......	0.0075"	0.015"	......
0.1	5.6	......	......	0.0725"	......	......	......	0.016"
0.11	5.86	......	0.0425"	......	......	......	......	......
0.2	7.68	......	......	......	0.0175"	......	......	......
0.3	9.34	......	......	......	......	0.02"	......	......
0.31	9.46	0.032"	......	......	......	......	......	......
0.33	9.76	......	......	......	......	......	......	0.0375"
0.35	10.05	......	......	......	......	......	0.0375"	......
0.37	10.3	......	0.090"	......	......	......	......	......
0.4	10.7	......	......	0.1675"	......	......	......	......
0.41	10.8	......	......	......	0.025"	0.025"	......	......
0.5	11.96	......	......	......	......	......	......	......
0.55	12.5	......	......	......	......	......	0.0475"	......
0.65	13.6	......	......	......	......	......	......	......
0.66	13.7	......	0.135"	......	......	......	......	......
0.71	14.25	......	......	......	0.035"	......	......	......
0.76	14.7	0.058"	......	......	......	......	0.055"	......
0.775	14.85	......	......	......	......	......	......	0.06"
0.8	15.1	0.06"	......	0.25"	......	......	......	......
0.81	15.2	......	......	......	......	0.04"	......	......
1.0	16.88	......	......	......	......	......	......	......
1.05	17.3	......	......	......	0.0475"	......	......	......
1.10	17.7	......	......	......	......	......	......	0.075"
1.15	18.1	......	......	......	......	......	0.07"	......
1.22	18.66	......	......	0.3125"	......	......	......	......
1.35	19.66	......	......	......	......	......	......	......
1.45	20.36	......	......	......	......	0.0575"	......	......
1.5	20.9	......	......	......	......	......	......	......
1.55	21.05	......	......	......	......	......	0.0825"	......
1.6	21.4	0.085"	......	......	0.06"	......	......	0.09"
1.61	21.45	......	0.2175"	......	......	......	......	......
1.75	22.36	......	......	0.3825"	......	......	......	......
1.925	23.45	......	......	......	......	0.0625"	......	......
2.0	23.9	......	......	0.42"	......	0.07"	......	......
2.1	24.5	......	......	......	0.0725"	......	......	0.105"
2.2	25.1	......	0.2525"	......	......	......	0.1"	......
2.25	25.35	......	......	......	......	0.075"	......	......
2.45	26.45	0.11"	......	......	......	......	......	......

TABLE 4. RESISTANCE IN IN. OF WATER OF AIR FLOW THROUGH COMPOSITE FILTERS—A TESTS

Vel. press. through ⅝" noz.	Vel. through material	Cotton flannel	3/16" Cotton held by Cotton flannel	3/16" Cotton between muslin
		.0335 lb. per sq. ft.	water = 0.146 lb. per sq. ft.	
0.09	5.4	......	......	0.08
0.11	5.86	0.01	0.085	......
0.3	9.34	0.0175	......	......
0.31	9.46	......	0.1875	......
0.5	11.96	0.0225	......	......
0.65	13.6	0.0275	......	......
0.8	15.1	......	0.3225	0.2725
1.0	16.88	0.035	......	......
1.15	18.1	......	0.40	......
1.35	19.66	0.0425	......	......
1.5	20.7	......	......	0.3875
1.6	21.4	......	0.4875	......
1.8	22.7	0.0475	......	......
1.85	23.0	......	0.5325	......
1.9	23.3	......	......	0.44
2.45	26.45	0.06	......	......

TABLE 5. CALIBRATION OF NOZZLES, USED IN FILTER TESTS, WITH PLAIN ORIFICES

Coefficients used are 1.0 for nozzles and 0.60 for orifices

5/8" Nozzle		7/8" Orifice		31/32" Nozzle		1 3/16" Orifice	
V.P.	A.P.M.	S.P. loss	A.P.M.	V.P.	A.P.M.	S.P. loss	A.P.M.
0.005	0.603	0.018	1.34	0.000	0.00	0.034	3.4
0.008	0.763	0.021	1.45	0.005	1.45	0.047	3.975
0.011	0.895	0.023	1.52	0.013	2.36	0.052	4.2
0.018	1.14	0.028	1.67	0.04	4.09	0.082	5.26
0.038	1.66	0.040	2.0	0.063	5.15	0.11	6.1
0.053	1.96	0.048	2.19	0.12	7.06	0.188	7.96
0.068	2.23	0.059	2.31	0.148	7.86	0.203	8.3
0.091	2.58	0.076	2.77	0.21	9.34	0.268	9.54
0.126	3.03	0.098	3.14	0.275	10.71	0.345	10.8
0.146	3.26	0.111	3.34	0.34	11.9	0.418	11.88
0.228	4.07	0.164	4.06	0.50	14.4	0.593	14.2
0.268	4.42	0.198	4.46	0.635	14.9	0.748	15.95
0.416	5.5	0.296	5.35	0.732	17.48	0.858	17.2
0.55	6.32	0.388	6.24	0.815	18.45	0.958	18.2
0.713	7.2	0.505	7.12				
1.028	8.63	0.728	8.55				
1.133	9.06	0.798	8.95				
1.248	9.52	0.885	9.44				
1.34	9.86	0.958	9.8				

Filter Area = 9" x 8" = 72 sq. in. = 0.5 sq. ft.
Velocity through filter = 2 x A.P.M.

TABLE 6. STATIC PRESSURE LOSS—THROUGH MATERIAL

Vel. press. through 31/32" Nozzle	Vel. through Material	1 thickness China Silk		3/4" Absorbent Cotton through China Silk	Copper Wire Screen 200 Mesh
		Clean No. 8	dirty No. 9	No. 10	No. 14
0.01	8.0	......	......	0.085	......
0.012	8.4	......	0.06	......	......
0.025	9.4	......	......	0.14	......
0.04	11.0	......	0.118	......	......
0.055	11.64	0.027	......	......	......
0.06	11.8	......	0.148	0.195	......
0.07	12.4	0.03	......	......	......
0.095	13.9	......	......	......	0.004
0.1	14.2	0.035	0.183	0.227	......
0.16	17.1	......	......	......	0.005
0.2	18.8	0.046	0.27	0.301	0.006
0.3	22.4	0.056	0.348	0.357	......
0.4	25.5	......	......	0.41	0.008
0.5	28.4	......	0.48	......	......
0.6	31.0	0.06	......	0.5	0.01
0.7	33.6	0.088	0.589	......	......
0.8	36.0	0.095	......	0.578	0.012
0.9	38.2	0.102	0.686	......	......
1.00	40.3	0.107	......	0.647	0.013
1.04	41.2	......	0.174	......	......
1.2	44.1	0.119	......	0.711	0.014
1.38	47.2	......	0.933	......	......
1.4	47.5	0.13	......	0.77	0.015
1.6	50.7	0.142	0.978	0.827	0.016
1.8	53.8	0.152	......	0.888	0.0165
		0.00846 lb. per sq. ft.		wt. cotton = 0.0972 lb. per sq. ft.	

TABLE 7. STATIC PRESSURE LOSS—THROUGH FELT—B TESTS

Vel. Pres. through 31/32" nozzle	Vel. through felt	No. 1 1/8" C	No. 2 3/16" C	No. 3 5/16" C	No. 4 1/16" wool	No. 5 1/8" wool 0.146 lb. pr. sq. ft.	No. 6 3/16" wool	No. 7 1/8" Mix
0.005	7.6	.....	.....	.....	0.025	.....	.....	.....
0.02	8.9	0.044	0.145	.....	.....	0.094	0.094	0.101
0.03	9.7	.....	.....	.....	0.038	.....	.....	.....
0.04	10.5	0.048	0.159	0.293	.....	0.108	0.105	.....
0.06	11.9	0.054	0.176	0.317	0.046	0.117	0.115	0.134
0.1	14.2	0.061	0.205	0.372	0.055	0.137	0.134	0.16
0.2	18.8	0.079	0.264	0.483	0.074	0.178	0.174	0.219
0.3	22.4	0.094	0.311	0.57	0.088	0.213	0.207	0.252
0.4	25.5	0.105	0.35	0.645	0.1	0.243	0.234	0.285
0.6	31.0	0.125	0.425	0.77	0.121	0.291	0.28	0.344
0.8	36.0	0.144	0.486	0.885	0.139	0.33	0.32	0.396
0.9	38.2	.....	.....	0.94	.....	.....	.....	.....
1.0	40.3	0.159	0.541	.....	0.155	0.369	0.356	0.443
1.2	44.1	0.175	0.592	.....	0.169	0.404	0.393	0.486
1.4	47.5	0.188	0.636	.....	0.184	0.437	0.424	0.525
1.6	50.8	0.201	0.68	.....	0.196	0.467	0.453	0.562
1.8	53.8	0.213	0.719	.....	0.208	0.492	0.483	0.596
Wt. in lb. per sq. ft.		0.053	0.1255	0.2265	0.057	0.146	0.156	0.092

TABLE 8. PRESSURE LOSS EQUATIONS FOR THE MATERIALS TESTED

1 — A; R = 0.0046 Vm
2 — A; R = 0.0111 Vm
3 — A; R = 0.019 Vm
4 — A; R = 0.0032 Vm
5 — A; R = 0.0032 Vm
6 — A; R = 0.0041 Vm
7 — A; R = 0.00475 Vm
8 — B; R = 0.0026 Vm
9 — B; R = 0.0144 V = 00004 V^2
10 — B; R = 0.0161 Vm
14 — R = 0.0032 Vm

1 — B; R = 0.00375 Vm
2 — B; R = 0.013 Vm
3 — B; R = 0.0236 Vm
4 — B; R = 0.00385 Vm
5 — B; R = 0.00890 Vm
6 — B; R = 0.00855 Vm
7 — B; R = 0.01085 Vm
11 — A; R = 0.0026 Vm
12 — A; R = 0.0258 Vm
13 — A; R = 0.0200 Vm

Approximate average formulas for materials tested in terms of resistance loss for 1 lb. per sq. ft. and 1 ft. per min. vel. through material.

Let R = resistance loss; W = weight per sq. ft. and Vm = vel. through material.

For wool felt.
R = 0.052 W.V.

For cotton felt.
R = 0.0985 W.V.

For cotton batting.
R = 0.1205 W.V.

For hard felt composition.
R = 0.115 W.V.

No. 602

BY-PRODUCT COKE OVENS AND THEIR RELATION TO OUR FUEL SUPPLY

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Non-Member

THIS paper is intended to deal specifically with the by-product coke industry and its relation to conservation of our national fuel resources. It is desirable to review, as briefly as possible, the statistics covering the broad subject of fuel supply. We can get a better idea of the relative importance of the coke industry if we keep in mind certain facts about the fuel resources of the world.

The fuels upon which we are chiefly dependent are: coal, petroleum, natural gas and wood.

COAL RESERVE

In 1910 Van Hise wrote as follows: "Coal is by far the most important of all the mineral products. Next to coal in importance is iron. These two are of much greater consequence than all the other mineral products put together. The existence of extensive coal and iron fields has profoundly influenced modern civilization. The greatest commercial nations are America, England and Germany, and each has extensive coal and iron deposits. Little Belgium, because of its important coal and iron deposits, is a hive of industry, occupying a position as a manufacturing nation far beyond what one would expect from its limited area and population."

The truth in this statement has been clearly demonstrated during the past ten years. Coal is important, not only because it is essential to commercial and military survival, but also because it is the supply from which we will have to draw more heavily in the future.

The coal reserve of the world has been estimated to exceed 8,000 billion tons. This enormous reserve is subdivided as indicated in Table 1.

Consideration of these figures makes it quite clear that the United States, controlling more than half the world's supply of coal, is in a very favorable commercial position. It should be our pleasant duty as engineers to do what we can to maintain this tremendous asset.

Some of this coal reserve lies below practical mining depth, at least according to present practice. It has been estimated that nearly 85 per cent

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Paper presented originally before Michigan Chapter; also at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, June, 1921.

TABLE 1. DISTRIBUTION OF COAL RESERVE OF THE WORLD

Nation	Percentage of Total
United States, including Alaska	52.0
Canada	16.6
China	13.5
Germany	5.7
Great Britain and Ireland	2.6
Siberia	2.4
Australia	2.2
India	1.0
Russia in Europe	.8
Other Countries	3.2
	100.0

of the coal reserve of the United States lies at a depth not to exceed 3,000 ft. An idea of the volume of this 85 per cent is gained from Professor Fernald's statement that if all this coal be placed in one cubical pile as solid as the coal in the ground, the pile would be 8 miles long, 8 miles wide and 8 miles high. It is estimated that since we have been mining coal in the past century, we have exhausted only ½ of 1 per cent of our coal reserve. This means a thin slice, (about 200 ft. thick) from our 8 mile cube.

It is an interesting pastime to speculate as to the number of years we will pass before our coal supply is exhausted and each man is entitled to his own guess. The flat statement that we have used only ½ of 1 per cent of the total available supply might allay our fears, but there are certain other real facts that must be kept in mind:

1. Practically all the high-grade coal is in the eastern half of the country and the volume of this high-grade coal amounts approximately to but 15 per cent of the total.
2. About 70 per cent of the entire coal reserve is low-grade bituminous or lignite.
3. The development of the nation's waterfall, when completed so far as is considered practical, will not supply the energy we now use—we cannot depend on water power to carry us when the supply of mineral fuel is exhausted.
4. Our rate of consumption of coal is rapidly increasing—since large figures are bewildering, this can best be shown by the following table of per capita coal consumption:

1870	0.96 short tons per capita
1880	1.4 " "
1890	2.3 " "
1900	3.2 " "
1915	5.5 " "

The per capita consumption has increased not only because of the shift of population from the country to the city, but also because of the more general use of coal as a domestic fuel by farmers. Wood is scarce and farm labor is scarce, so a great many farmers have broken away from the old practice of using wood exclusively as fuel. This probably does not mean a tremendous increase in the total consumption, but is a factor that is usually overlooked by those of us who are interested chiefly in industrial life.

At the present time the production of coal is less than the demand and it has been estimated that the demand for the present year is in the neighborhood of one billion tons. Looking some distance in the future we can see that, foreign coal fields will probably be exhausted before ours and that export requirements will cause rapid acceleration of coal production in America.

The consumption of coal will be increased not only by the depletion of our forests and the increasing intensity of industrial life, but also by exhaustion of our natural gas fields. Most of us will live to see the time when there will be no natural gas available for industrial use. The consumption of natural gas in 1915 was estimated at 629,000,000,000 cu. ft. and we can easily figure that it will require 26,000,000 tons of coal annually to replace this volume of gas.

PETROLEUM

The total world output of petroleum up to 1917 has been estimated at 7,000,000,000 barrels of 42 gal. each and of this the United States has produced approximately 4,000,000,000 barrels, or about 57 per cent. More recently the annual production of the United States has been more than two-thirds of the world production. The amount of oil in the known and prospective oil fields in 1916, in the United States, has been estimated at 7,000,000,000 barrels and this reserve will be exhausted in about 25 years.

This does not mean that 25 years from now we will be entirely out of oil, but it does mean that from now on the production of oil will be increasingly difficult and that production from fields now pumped will soon have to be supplemented by the extraction of oil from shales. These oil bearing shales have been investigated to some extent by the Bureau of Mines and U. S. Geological Survey and it has been estimated that there is an area of 3000 square miles in the Colorado-Utah district which is underlaid by shale beds, from which it will be possible to extract more than 150,000,000,000 barrels of oil, or more than twenty times the amount originally available from wells. The conclusion to be drawn from these statements is that oil obtained from now on will be relatively more expensive and this in turn proves that it is of importance that as much oil as possible be recovered from coals through by-product processes.

The bulk of the present petroleum production is now converted into gasoline and that is why a real substitute for gasoline is so eagerly sought.

NATURAL GAS

So far as the writer knows, no one has made definite figures as to the volume of natural gas in the ground. The first gas was used in 1821 and the rate of consumption has increased very rapidly in recent years. Some of our large gas fields have been exhausted and there is a continual decrease in pressure in existing fields. It seems to be agreed that the cream has been skimmed and according to some estimates, the supply will be practically exhausted within the next 20 years.

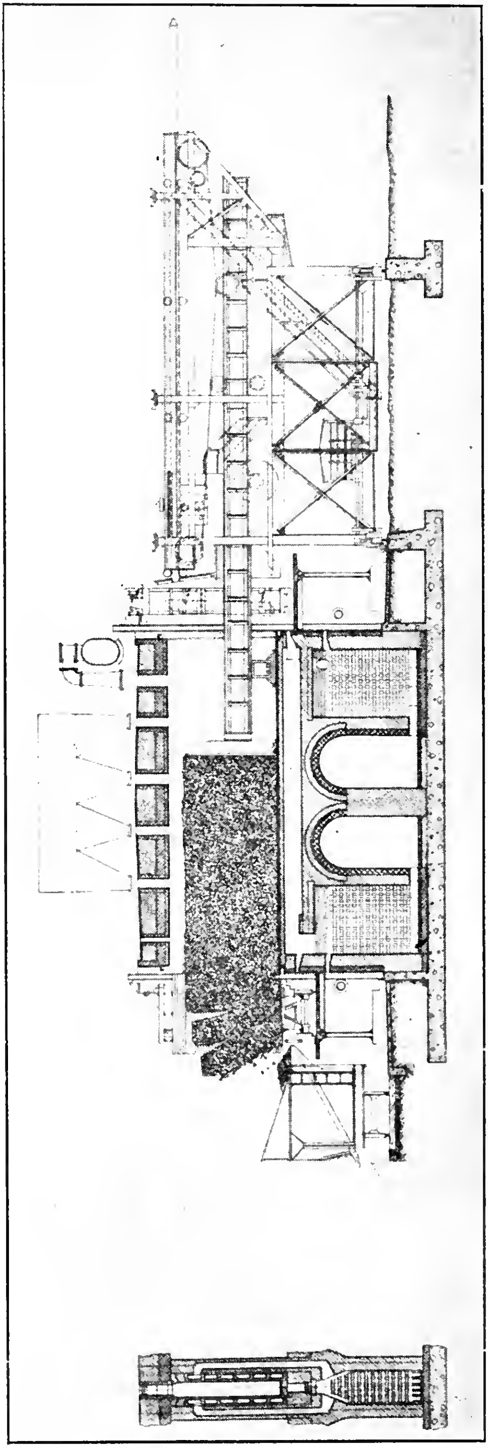

FIG. 1. SECTIONAL VIEW SHOWING ARRANGEMENT OF ELEMENTS OF REGENERATIVE COKE OVENS

WOOD

Figures on the original wood supply are not at hand, but it seems safe to say that more than half of it has been used, and that we are cutting forests at a rate three times as fast as timber is being grown. Only a few years ago, the timber supply was considered inexhaustible by our fathers and it was wasted outrageously. The best that we can say is that the timber now standing is a fuel resource of only moderate value.

POSSIBLE CONSERVATION OF FUEL

We come now to the interesting study of what can be done to prolong the fuel supply without at the same time interrupting commercial development. Since coal is the fuel upon which we must rely in the future, it should naturally be given most attention. In order that we may discuss possible savings intelligently, we should first find out what becomes of the annual production of coal. The following table taken from an article by David M. Myers in the magazine *Combustion,* gives a subdivision of the 1918 consumption of 684,000,000 tons:

TABLE 2. DISTRIBUTION OF COAL CONSUMPTION FOR YEAR 1918

	Percent.
Railroads	28.0
Small and steam trade	4.0
Industrial	33.0
Steam at the mines	2.0
Steamship bunker fuel—Tidewater	2.0
Steamship bunker fuel—Great Lakes	0.3
Bee hive coke	9.3
By-product coke	4.3
Coal gas	1.0
Exported	4.0
Special uses	0.1
Domestic heating	12.0
	100.0

Since these figures are in the form of percentages, they will be approximately correct for the current year although we know that the ratio of bee-hive to by-product coke has changed and it is probable that our exports have increased.

Perhaps it is not best to be alarmists but we must not grow complacent over a mental picture of a block of coal 8 miles long by 8 miles wide by 8 miles high. The figures on coal in the ground may be misleading and when comparing them with annual production, we must take into account the coal wasted by common mining practice. George Otis Smith, Director of the U. S. Geological Survey, stated before a recent meeting of the American Iron and Steel Institute: "That of a ton of coal in place, where nature stored it for the use of man, the amount converted into mechanical energy, under the average practice of today, is only 76 pounds." If only 4 per cent of the coal is to be used effectively, the reserve will not last forever.

We will do well, therefore, to keep the fuel situation in mind and pledge ourselves to help the cause of conservation, at least whenever we can do so without cost to ourselves. In solving our daily problems we should ask ourselves if a little more thought will not enable us to get the desired results with less fuel. There have been countless suggestions for saving fuel and no doubt most of these suggestions are good ones. Certainly it is necessary to rigorously follow up every possibility if we are to approach ideal conditions.

However, for purposes of comparison we can classify some of the methods that obviously will materially affect the total fuel consumption:

1. Extension of by-product coking process to handle greater percentages of the total coal used;
2. Development of the nation's water power;
3. Electrification of the steam railroads;
4. Utilization of low-grade fuels (such as anthracite culm, bone coal from bituminous mines, coke breeze, lignite, etc.) which in the past have been wasted or neglected;
5. Use of solar heat.

Use of Solar Heat.—Taking up the last item first, it will be agreed that the commercial development of heat from the sun's rays is impractical at present. It may be of interest, however, to note that Dr. Steinmetz has estimated that if all the heat from the sunshine available in the United States is used, the total will be more than 800 times as great as the energy from all water power theoretically available for development.

Utilization of Low Grade Fuel.—In regard to the use of low grade fuels, it may be said that rapid developments have taken place in the last few years, chiefly in the use of chain grates and in pulverized-fuel burners. It is safe to say that 2 per cent of the total solid fuel produced in the past has been discarded because we did not know how to use it, and it seems safe to say that within a few years this loss will be eliminated. With an annual production of 800,000,000 tons of coal, such a saving will amount to 16,000,000 tons per year. Anthracite culm, or No. 4 buckwheat, and coke breeze are now burned very nicely. The writer knows of no place where bone coal is handled successfully by mechanical stokers, but experiments have been made and no doubt the problem will eventually be solved. Remarkable results have been reported from installations where low-grade bituminous coal is pulverized.

Electrification of Steam Railroads.—The above mentioned estimate charges the steam railroads with the use of 28 per cent of the total annual production of coal. It has been estimated that it takes 7 lb. of coal to generate the equivalent of one kilowatt-hour when burned in locomotives, whereas the same amount of power can be produced in a modern power house with a consumption of only 2.5 lb. of coal. If all steam railroads are electrified (and this will no doubt be done in this century), this fuel item will be reduced in the ratio of 7 to 2.5, or to about 36 per cent. This will bring the coal consumption chargeable to railroads down to 10 per cent of the total coal consumed in the nation. Assuming the 1920 consumption to be 800,000,000 tons, the reduction due to electrification of steam railroads would have been 144,000,000 tons and the actual consumption would have been only 80,000,000 tons.

Utilization of Water Power.—Development of the water powers in the United States is receiving increased attention and new construction is bound to be accelerated by the law creating a Federal Power Commission, which was signed June 11, 1920. It has been estimated that we have developed not over 10,000,000 h.p. of a possible 50,000,000 h.p. In other words we have developed only 20 per cent of the total potential water power.

Assuming an average overall efficiency in generators of 90 per cent, 10,000,000 h.p. is equivalent to 6,700,000 k.w. Assuming again 2½ lb. coal per kw.-h., we can figure that water power now developed is saving

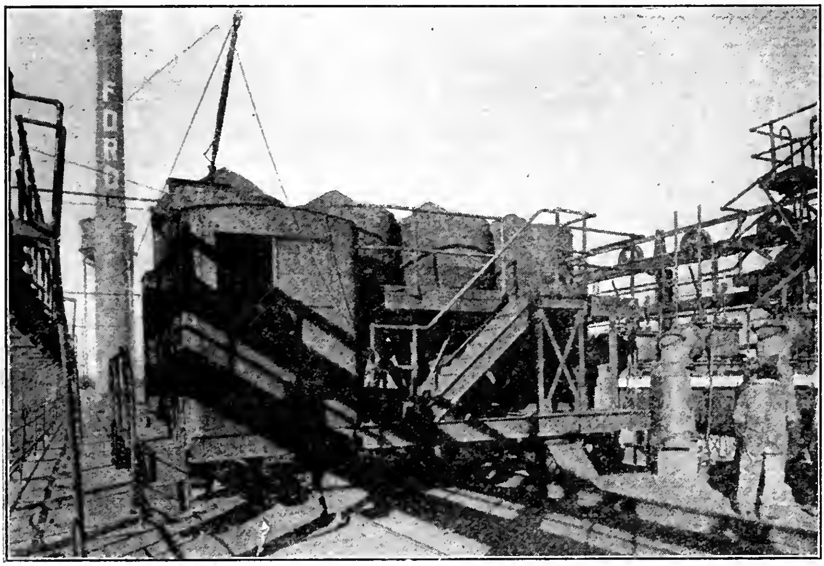

FIG. 2. VIEW OF COAL CHARGING CAR

the nation coal at the rate of 73,000,000 tons per year. When the total potential waterfall of 50,000,000 h.p. is developed, the annual saving in coal would be 365,000,000 tons, or about one-half our present consumption.

Extension of By-Product Coking Process.—The reason for placing by-product coking process first in the classification of the major methods of fuel conservation, is because the possibilities of this method are practically unlimited. The amount of coal that can be saved by development of water power, while an enormous item in itself, is practically fixed. But the fuel that can be saved by by-product operations will be an increasing percentage of the total consumption. Moreover, it is probable that we may soon find that the only commercially practicable way to use the relatively great quantity of lower-grade coal is by means of some coking process.

Marked progress has been made in recent years in coking coals hitherto considered in the non-coking class. No one knows precisely what quality makes coal coke, but we are hopeful that the next decade will see this mystery cleared up and that eventually all bituminous coals will be brought into the coking class. It is not at all unreasonable to hope that in this way, the low-grade coals of Indiana and Illinois will be converted into very satisfactory fuel and that this conversion will be attended by a recovery of by-products that will mean much to the country.

The by-product coke men have looked forward to the day when all the coke made will be produced in by-product ovens. That day is in sight. Even now they produce more than half the total coke made, and the war has so emphasized the advantages of this method that it is doubtful if many more bee-hive ovens will be built. It is probable that within the next 10 years, there will be enough capacity in by-product ovens to produce the coke required for metallurgical purposes.

Referring again to the subdivision of the 1918 coal consumption according to uses, it seems reasonable that in the near future the following classes will be handled in by-product ovens: Bee-hive coke (9.3 per cent); by-product coke (4.3 per cent); coal gas (1.0 per cent); domestic heating (12.0 per cent); total 26.6 per cent. Applying this percentage to an annual production of 1,000,000,000 tons, we will then be handling 266,-000,000 tons. Now let us consider the effect of coking this amount of coal in the terms of by-products recovered.

Ammonia is valuable, not only for refrigerating purposes, but also as a fertilizer, because the nitrogen it contains is in an available form. More than 80 per cent of ammonia, by weight, is nitrogen.

Whenever bituminous coal is burned in its raw state in any kind of furnaces, the ammonia is driven off into the atmosphere and wasted. True, we now have plants for extracting nitrogen from air, but it is not logical to drive the ammonia into the air and then recover it at great expense, when it can be recovered directly from coal in the by-product coking process. We can assume a yield of 20 lb. of ammonia (in the form of sulphate) per ton of coal. The annual recovery from 266,000,000 tons will therefore be 2,660,000 tons. Assuming that crop fertilizer will contain 8 per cent ammonium sulphate, we can figure that we have enough for 32,500,000 tons of fertilizer. Further assuming 100 lb. of fertilizer per acre, we can figure that we have enough fertilizer to enrich 650,000,000 acres each year. This is an area nearly four times as large as the State of Texas.

Coke oven gas is quite satisfactory for use anywhere that gaseous fuel is desired. Gas is a refined fuel and can be used for heat treatment of metals where solid fuels will not serve. It can be burned under boilers at from 75 to 85 per cent over-all efficiency, but at present, owing to the high efficiency that can be obtained with low-grade solid fuels on modern stokers, it is considered best to reserve the gas for use in the metallurgical industries and for domestic purposes. It is better adapted to industrial work than producer gas, not only because of its higher heat value, but also because it is cleaner and more uniform in quality. Its use is bound to be extended in the future and no doubt most industries now served by natural gas, will have to turn to coke-oven gas as a fuel. Due to its lower CO content, it is less poisonous than other forms of artificial gas.

Assuming a yield of 6000 cu. ft. of surplus gas per ton of coal, 266,-000,000 tons of coal will yield over 4,300,000,000 cu. ft. per day. This gas would supply 65 cities as large as Chicago. Incidentally, it seems that plans now under way at Chicago, contemplate the use of gas from by-product ovens as the sole supply.

Conservation of Petroleum Due to Extended Use of Motor Benzol.—The light oil or crude benzol recovered by scrubbing the gas, forms a source for many valuable compounds. During the war the toluol and

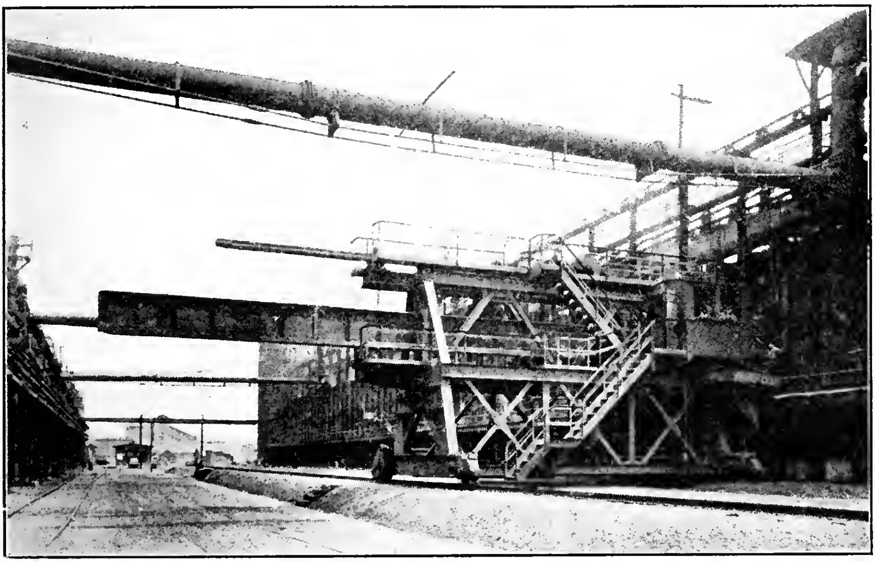

FIG. 3. VIEW OF COKE PUSHER IN OPERATION

benzol derived mainly from this source was converted into TNT and picric acid. In peace times no doubt most of it will be converted into motor benzol.

Most motorists claim that gasoline quality has declined; in any event benzol is a much superior motor fuel at present. A mixture of equal parts of benzol and gasoline will give from 20 to 25 per cent more mileage to the gallon than straight gasoline. The pressure due to the explosion of such a mixture in the engine cylinder is maintained through a greater percentage of the stroke. For this reason it is found that benzol-gasoline mixtures make it much easier for cars to climb hills.

It is anticipated that if denatured alcohol ever becomes a popular substitute for gasoline it will be due to the quality of benzol. It is claimed that alcohol does not work well alone and will not mix with gasoline. It is readily miscible with benzol and the mixture gives good results.

We may assume a yield of 2.5 gal. of motor benzol per ton of coal coked. The annual yield from 266,000,000 tons of coal that we hope to coke in by-product ovens, may therefore be estimated at 650,000,000 gal. Assuming that the average automobile travels 5000 miles per year and will travel 18 miles on a gallon of benzol, this average car will therefore consume 278 gal. The quantity of benzol thus recovered would therefore supply 2,350,000 automobiles with the necessary fuel.

CONSTRUCTION OF COKE OVENS

The first coke was made in the United States by covering with earth a pile of coal which had previously been ignited. This was a natural adaption of the older processes followed in the production of charcoal. The first bee-hive ovens were built in the Connellsville, Pa., region in 1841. Round chambers with dome shaped roofs of stone and fire brick were used. In starting this process, coal was burned in the chamber to heat up the inside surface. Then a charge of coal was dropped into the chamber and the air supply so regulated that the volatile matter (which was driven off by the radiant heat from the roof), was burned and in this way a coking temperature was maintained. After a period of from 30 to 72 hours the coke was quenched by water in the oven and then drawn out for sorting. In these ovens all the by-products were wasted and perhaps not more than 65 per cent of the coal was recovered in the form of coke.

The first battery of by-product ovens was built in Syracuse in 1892, Since that time there has been the rapid expansion that is typical of American industries.

In principle, a by-product oven is simply a retort with provision for external heating and for the collection of the volatile matters driven off. In modern practice, this retort is called the oven chamber. It is 35 to 40 ft. long, 10 to 12 ft. high, and 16 in. to 20 in. wide. The heating flues on either side of the oven chamber are built of silica brick and are heated by gas. Ovens have been built in batteries or blocks of as many as 120 ovens, although 60 ovens per block is near the present average.

Two general types of by-product ovens have been built, the recuperative and the regenerative. In the recuperative type, the gases in the heating flues flow continuously in one direction, the products of combustion being used to preheat the air necessary for combustion in the oven flues. The waste gas is passed out through a flue or flues around which the incoming air is made to flow. The waste gas from each oven then passes into a main flue parallel to the oven block and thence through waste-heat boilers to the stack.

In recent years the regenerative type has gained in favor and practically all the ovens now under construction are of this type. In this type there are two main flues and two sections of checker brick. The products of combustion pass out through one set of checker-brick chambers to one main flue, while air for the oven-heating flues is drawn in through the other main flue and checker-work chambers. The flow of gas and air is reversed at frequent intervals (usually at least twice per hour), thus allowing the incoming air to take up the heat absorbed by the checker brick from the waste gas.

Recuperative and regenerative types may have either horizontal or vertical flues for heating the oven chambers and the gas may pass through the flues in either series or parallel paths.

OPERATION OF BY-PRODUCT COKE OVENS

Receiving and Unloading Coal.—Coal may be brought into the plant by rail or water. Rail coal may come in hopper-bottom cars from which it may be dropped into track hoppers. The coal may be fed from the track hoppers onto a conveyor belt which may carry it to the coal preparation plant or to storage. Rail coal may also be handled by means of a car dumper, which lifts the car bodily from the track and turns its con-

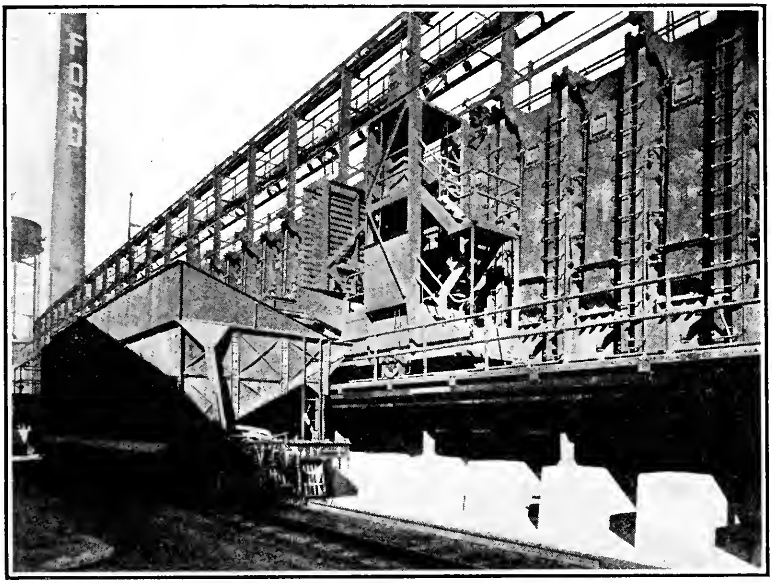

FIG. 4. VIEW OF COKE GUIDE AND QUENCHING CAR

tents into a hopper. The coal from this hopper may be fed onto a conveyor belt which may carry it to the preparation plant, or to storage, or it may be fed into a transfer car to be carried to storage.

Coal brought to the Semet-Solvay plant by Great Lakes freighters, which have 6000 to 13,000 tons capacity, is usually lifted out of the ship's hold with electrically-operated unloading towers. These towers have been built with a maximum capacity of 1000 tons per hour from boat to hopper on the unloader. From the hopper on the unloader, the coal is fed onto a conveyor belt, or into transfer cars to be taken to the coal preparation plant, or to storage.

Coal Storage.—Some plants store 200,000 or 300,000 tons of coal during the season of navigation on the lakes. Smaller plants store coal by dumping it from hopper-bottom cars spotted on a trestle. This coal may be reclaimed by use of a locomotive crane. Larger plants use a coal bridge, similar to the ore bridge, seen at most blast-furnace plants. Coal may be fed onto the bridge by a belt conveyor, or it may be dumped into a circular or rectangular receiving pit, from which it is lifted and carried to the storage pile by a clam-shell bucket of 4 to 10 tons capacity. The bridge structure usually supports a hopper into which the coal may be dipped by the clam-shell bucket when the bridge is used for reclaiming. The coal may be taken from the bridge hopper by a transfer car or belt conveyor.

Coal Preparation.—In the preparation plant, the coal passes through a rough crusher or a Bradford breaker, where it is reduced in size so that 90 per cent will pass through a 1¼ in. square opening. From the rough crusher, the coal is elevated to the mixing bins, whence it passes through a proportional feeder into the pulverizer or hammer mill. Most plants use a mixture of high- and low-volative coals, so that the proportional feeder is necessary to insure the correct percentage of each. The hammer mill grinds the coal to a fineness such that from 60 to 95 per cent of it will pass through a screen with ⅛ in. square openings. This fine coal is elevated and conveyed to the storage bins over the ovens.

Charging the Ovens.—After an oven has been pushed, the doors at the end of the chamber are closed and sealed with luting clay. The coal is taken from the storage bins, which are located over the oven block, in an electrically-operated charging car, to a point directly over the oven. In charging, 12 to 16 tons of coal are dumped into the oven, an even distribution being obtained by the horizontal reciprocating motion of a leveling bar. This leveling bar is usually mounted on the pusher. The leveling process is necessary in order that the maximum tonnage of coal may be charged into the ovens; it also insures a free passage of the gas out of the oven. After the coal is dropped into the oven, the charging hole covers are closed and the leveling doors at the end of the oven are closed.

Process of Coking.—The coal is sealed in the oven for from 12 to 20 hr. (depending on the type and width of oven, the quality of coal charged, and the quality of coke desired), the gas being driven off through the vertical riser into the hydraulic main. The gas driven off during the first half of the coking period is richer in illuminants and higher in calorific value; and, where gas of high calorific value is supplied for domestic use, it is the custom to have a double collector pipe on the oven block so that the *rich* gas can be separated from the *lean,* the latter being used to heat the ovens. Where the surplus gas is not sold for domestic purposes, the separation of the gas is usually unnecessary and the surplus gas is then utilized for industrial purposes, or lacking a market, it is burned under boilers in the plant.

The temperature in the oven chamber or retort is maintained by gas burned in the flues on each side of it; a temperature of from 1000 to 15,000 deg. cent. is carried in these flues.

Exhausting and Scrubbing the Gas.—As the gas leaves the oven, it passes through the collector pipe (or hydraulic main) where it is either showered with water and weak ammonia liquor, or the pipe may be flushed with tar circulation, to prevent stoppages. The former method throws down considerable tar and ammonia, these being separated by decantation. After leaving the collector pipe, the gas is drawn through a cooler where it is again sprayed with weak liquor and more tar and ammonia are removed. The gas is drawn through this cooler by an exhauster of either the positive or centrifugal type. This exhauster assists in the removal of the tar from the gas. The suction on the exhauster is usually about 4 to 8 in. water gage and the discharge pressure is seldom more than 3½ lb., depending upon the scrubbing equipment and the friction in the pipe lines. The temperature of the gas at the exhauster ranges from 25 to 50 deg. cent.

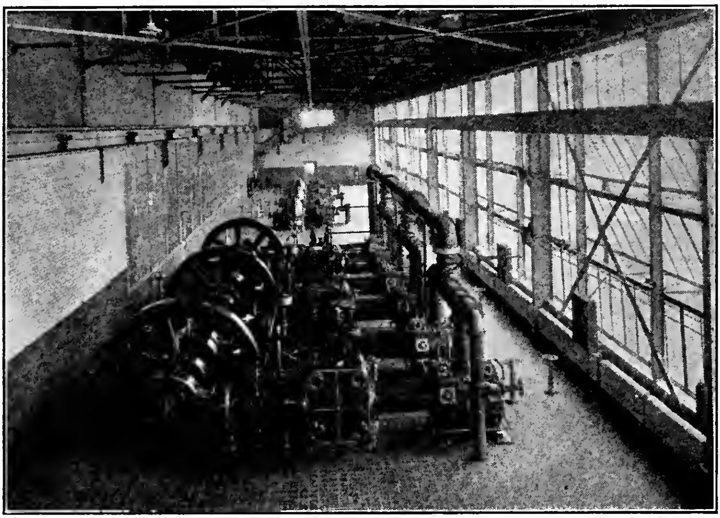

FIG. 5. BLOWING ENGINES FOR BY-PRODUCT COKE PLANT

The exhauster forces the gas through an ammonia scrubber of the counter-current type. Here the last of the ammonia is taken out and the gas then passes through a benzol scrubber where it is sprayed with an absorbing oil instead of water or weak liquor. It is then ready for commercial distribution without further treatment, except when it is to be used for domestic purposes, in which case it is purified from sulphur in purifiers of the type common in other gas manufacturing plants.

Ammonia liquor and tar, recovered from the circulation systems mentioned, are collected into large decanters and separated by their specific gravities. The moisture in the tar is driven off by heat and the tar is then ready for shipment to the refiner.

The ammonia is driven off from the weak liquor (containing 1½ to 2 per cent of ammonia) by heat in the still. The fixed ammonia (carbonate, chloride, etc.) is set free by treatment with lime. The ammonia vapor is absorbed in water and the product, containing some impurities, is the crude ammonia liquor of commerce.

The crude benzol is recovered from the absorbing oil by distillation, the absorbing oil being again returned to the scrubber.

Quenching the Coke.—After practically all the gas has been driven out of the coal in the retort, the doors are opened and the coke residue is pushed out into a quenching car by an electrically-operated ram. The quenching car is 40 to 50 ft. long and about 10 ft. wide, with a sloping bottom, and is usually drawn by an electric locomotive. While the coke is being pushed, the car is traversed in a direction parallel to the oven block at a speed of about 60 ft. per minute, so that the coke is distributed uniformly throughout its length. As soon as all the coke from the oven has been pushed into the car, the locomotive pushes it rapidly to the quenching station, where the red-hot coke is showered with water. Enough water is applied to promptly cool the outside of the coke and prevent combustion, but leaving enough heat in the center of the piece to drive off the excess water and so produce dry coke. The quenching car is then drawn to the coke wharf upon which the coke is discharged.

COKE PREPARATION

Because coke is more fragile than coal, its preparation requires more care. From the time it leaves the oven until it is delivered at the point of actual consumption, it should be handled carefully if best results are to be obtained.

The coke wharf is an inclined surface about 200 ft. long and is usually paved with cast-iron slabs. Cast iron is used in preference to bricks because coke slides over a cast-iron surface more readily. The angle of the slope is intended to be such as to cause the coke to slide readily onto the conveyor belt without jamming. This angle is commonly 28 deg. The coke is distributed evenly upon the belt which parallels the wharf, by means of a roll feeder. The conveyor belt elevates the coke to the first screen, which may be of the curved bar or the rotary-grizzly type. Foundry men generally want the larger lumps and where foundry coke is desired, the pieces passing over the first screen may be delivered to a belt, which loads the coke into the car to be consigned to the foundry. Large coke is also sometimes sent direct to the blast furnace, although in other cases the coke is crushed for furnace use and screened to a uniform egg size.

Various types of screens are used for the further sizing of domestic coke, the more common types being: rotary screen, grizzly screen, and ripple screen. The rotary and ripple screens have long been used in sizing all kinds of materials and are no doubt, familiar to all. The rotary grizzly is a later development; it is merely a number of cast-iron discs mounted on a series of shafts, all of which are driven by the same motor; discs on alternate shafts are staggered, thus leaving suitable openings for the undersize to pass through. The main idea of this screen is that it gives a mechanical movement to the coke in its passage across the screen and that the discs present a fresh surface. This screen should have a low maintenance cost as the discs can be replaced when worn.

When we get down to pea, buckwheat and breeze sizes, some form of vibrating screen seems to be most effective. This is because the fine dust constitutes a higher percentage of the mixture and since the dust is apt to be moist, the problem is to shake the fine particles free from the pea and buckwheat sizes.

The Mitchell screen is mounted on a frame work so that the screen sets at an angle such as will cause the coke to pass over it when the screen is vibrated. A small motor is mounted underneath the screen, with its bedplate rigidly secured to the frame work, and the motor shaft is extended at each end and operated in a special ball bearing which is secured to an iron member connected to the screen. The main motor bearings which are held in the motor end bells, are set slightly off center so each revolution of the motor imparts a vibration to the screen; this motor operates at 3600 r.p.m.

The Hummer screen is vibrated by solenoids which are supplied with 15 cycle alternating current; this screen therefore is vibrated due to the 1800 alterations per min. in the current supply. The Newaygo separator is an older design in which the screen is vibrated by taps from plungers which in turn are actuated by an eccentric on a driven shaft. The Robbins shaking screen is merely a screen of suitable mesh supported in a swing; a reciprocating movement is imparted to the screen by an eccentric which may be driven from a line shaft or from an individual motor.

YIELDS FROM ONE TON OF COAL

The yield of by-products from a ton of coal will naturally vary over a wide range, depending on the quality of coal, the rate of operation, and other important factors. For purposes of illustration, the following table is offered to show fair yields and market values:

Coke	.75 ton at $6.00	$4.50
Ammonium sulphate	25 lb. at 3c	.75
Motor benzol	2.5 gal. at 20c	.50
Tar	7 gal. at 5c	.35
Gas	6,000 cu. ft. at 15c	.90
Total gross return		$7.00

This gross return may be compared with the value of say $4.00 per ton for the original coal.

In forming conclusions from tabulations similar to the one given above, certain important facts must be kept in mind:

a. Construction of a by-product coke plant involves a large investment and fixed charges must be paid on this capital.

b. Conversion of coal into by-products is a large-scale operation; bulky materials are handled—labor, expert supervision, steam, water and electric power supply, maintenance—all these are important items of cost.

c. Wide fluctuation of costs and returns. For example, the return for gas varies from 5c. to 45c. per 1000 cu. ft. in various localities.

Economic laws work out in this as in any widely established industry. That is, any industrial plant properly designed and operated earns a fair average return on the capital invested. The real conclusion to be drawn from all the data that may be collected is that there are still many communities where a by-product coke industry can be profitably operated. In turn the benefits to the community may be summarized as follows:

A. Direct local benefit due to products of the plant made available, such as domestic and motor fuel. Gas can usually be sold at a lower figure than would otherwise be possible.

B. Direct local benefit, through employment of additional labor.

C. Indirect economic benefit due to conservation of national fuel supply.

D. Indirect benefit due to fact that other industries will be attracted by availability of superior fuels for special processes.

The development of the by-product coking process may be properly classed as one of the many logical steps in the advance of civilization, not only because it has fostered industrial progress, but because it conserves our greatest natural resources—a resource which economists agree has been woefully wasted, especially in this country.

COKE BREEZE

The better grades of coke naturally go to the iron and steel industry, or are used for domestic heating. The screenings from these grades is known as breeze. While this breeze contains plenty of combustible, it has not been an easy matter to teach firemen to handle it properly under a boiler and it is only recently that mechanical stokers have been properly designed to burn this fuel to advantage. The distinguishing features of these stokers are:

1. A chain grate with correct ratio of air to grate surface—really a moving pin hole grate.

2. An ignition arch to kindle the fire quickly in the green fuel, as it enters the furnace.

3. Design of the air-supply system so that the air pressure under the grate can be regulated to suit the conditions of the fuel bed.

Given coke breeze, all of which passes through ½ in. screen and which has a heating value of 11,500 B.t.u. and contains 2 per cent volatile, 18 to 20 per cent ash, and 12 per cent moisture, stoker manufacturers will guarantee to operate a boiler at 200 per cent of full-load rating with an over-all efficiency of 67 per cent.

Ten years ago this fuel was, for the most part, wasted. When we re member that perhaps the breeze represents 5 per cent of the coke yield, it is clear that its utilization is a distinct advance and is welcomed by the coke manufacturer, because it eliminates a waste which has been justly criticized in the past. Tests show it to be a much better fuel than No. 4 anthracite buckwheat, which now finds a ready market.

COKE AS A DOMESTIC FUEL

Perhaps the most popular reason for the use of coke is its cleanliness. The smoke nuisance is a question that has received much attention in recent years. The Mellon Institute has reported that the annual loss in the city of Pittsburg amounted to $10,000,000 or about $20 per capita. This is a big item—it must be about as large as the city's bill for domestic fuel.

It has been stated that in large cities in the soft coal region, 60 per cent

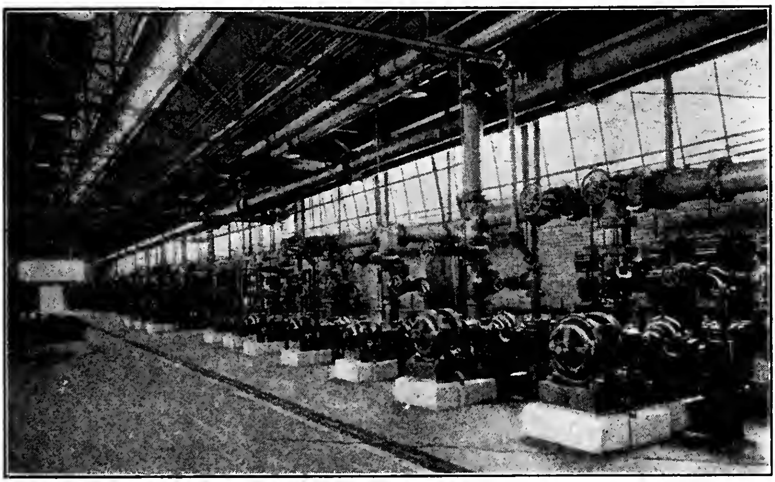

FIG. 6. VIEW SHOWING BY-PRODUCT PUMP ROOM

of all the smoke comes from house heating furnaces in the residential district. In other words the smoke originates in the district where it will do most harm to private property. This percentage tends to increase because with modern stokers, less smoke is produced by industrial plants. Electrification of railroad terminals also tends to reduce the smoke so that it becomes increasingly desirable to prevent the smoke from house-heating furnaces.

ELIMINATION OF SMOKE

The easy way to eliminate this smoke is to use a fuel that burns without smoke. In the past, the only solid smokeless fuel has been anthracite coal. The use of anthracite has been limited to the eastern section of the country because practically all of it comes from Pennsylvania. Higher freight rates and the knowledge that there is a definite limit to the volume of anthracite in the ground, insures increasing restriction of the area in which it will be used. This is indicated by the fact that in the east we are satisfied with a lower quality of anthracite. A great deal of anthracite runs from 18 to 24 per cent ash, whereas there is no difficulty found in producing coke that will run from 8 to 14 per cent ash.

Coke is the cleanest of domestic solid fuels. It is nearly dustless and such dust as it carries is of neutral color so that white materials are not smudged. When bituminous coal is burned, it forms a soot on the heating surfaces of the furnace and the removal of this soot is a disagreeable task. This soot is a heat insulator, so the furnace is efficient only when freshly cleaned. When coke is burned, no soot is formed and the efficiency of the furnace is therefore uniformly high.

Tests reported by the University of Illinois for a furnace of capacity to supply 800 sq. ft. of radiation show the relative efficiency of coke as indicated in Table 3.

TABLE 3. RELATIVE EFFICIENCY IN FURNACE OF COKE AND COAL

Fuel	Efficiency, percent.
By-product coke	61.63
Gas-house coke	56.22
Anthracite	51.93
Illinois coal (Williamson Co.)	48.00
Pocahontas coal	46.51

The efficiency shows the percentage of total heat actually utilized in heating the building, the boiler being operated at from 63 to 66 per cent of capacity under all cases.

It is true that coke will carry more ash than the coal from which it is made. This is illustrated by study of the analysis of coal and coke from an actual operation as shown in Table 4.

TABLE 4. ANALYSES OF COAL AND COKE FROM AN ACTUAL OPERATION

Analysis	Coal, percent.	Coke, percent.
Volatile	28.42	0.54
Fixed carbon	62.99	87.46
Ash	8.59	12.00
	100.00	100.00

The coke is the residue or solid matter of the coal and contains all the ash from the coal. The total solid matter in the coal is 71.58 per cent. The ash in the coke will be the ratio of the ash to the total solid in the coal $= 8.59 \div 71.58 = 12$ per cent. This is not a serious matter, however, because ash in bituminous coal runs so much lower than in anthracite that the ash in coke may readily be kept at a relatively low value.

Another advantage of coke, that is greatly appreciated by its users, is the fact that a furnace can be made to heat up very quickly. So far as the writer knows, there is no other solid fuel that is comparable with coke in this respect.

The universal use of coke as a domestic furnace fuel will have a beneficial effect on the community in that it will insure a greater production of gas. The lowest rate for artificial gas is found in cities where the gas is produced in by-product coke ovens. A big volume production is essential to a low production cost in the gas business, just as much as in other industries. A big volume cannot be economically produced unless there is an assured market for the coke. Viewed in this way the coke user not only has the satisfaction that should come from the knowledge that he is saving by-products that would otherwise be wasted, but also has the knowledge that by burning coke he is aiding in keeping down the cost of artificial gas.

The use of coke should tend to stabilize the supply of domestic fuel. The small coal dealer does not usually have facilities for a large storage of coal. The coke manufacturer as mentioned previously, frequently has a large stock of coal, and can thus produce coke during the winter, to supply the needs of those who have been unable to fill their fuel bins during the summer.

HANDLING COKE FIRE

The handling of a coke fire differs from the handling of a coal fire for two reasons:

1. Due to angularity of the pieces and to porosity of the structure, less draft is needed for a fire.
2. A cubic foot of anthracite coal weighs about 55 lb.; a cubic foot of bituminous coal weighs from 47 to 52 lb., whereas a cubic foot of coke weighs only 30 to 40 lb. Assuming the same efficiency for the three fuels and that 200 lb. of each is required to bank the furnace at night, then the fire box for anthracite needs to hold 3.6 cu. ft., for bituminous coal 4.0 cu. ft., and for coke 5.7 cu. ft.

At present in the East, at least, there is a handicap upon the coke user because most furnaces are equipped with fire pots none too big to burn anthracite properly. Therefore pains must be taken to get enough coke into a fire pot to permit proper control of draft and also proper banking of the fire at night. The relative volumes required in a fire pot for coke and coal, mentioned above, apply of course, to a furnace much larger than is found in the average house—the ratio of 5.7 to 3.6, or 157 per cent, should hold for smaller furnaces and it is the writer's belief that most of the increased volume should be obtained by designing the fire box for greater depth.

A great many homes are "built to sell," that is, the construction cost is held as low as is consistent with good appearance and it is the writer's belief that these homes are equipped with furnaces that are too small for best results with coke. It would seem that increasing the fire box 30 to 50 per cent would not greatly add to the cost of a home and it is sure that this would allow the owner to get better results with smokeless fuels. If heating and ventilating engineers will exert their influence in this direction they will be rendering a distinct service.

FIRING COKE

Five rules for burning coke in house heating furnaces have been formulated:

1. Carry a deep bed of fuel—an average bed 18 to 20 in. deep will give good results.
2. Use very little draft after the fire is started and keep draft ıer control at all times.
3. Do not stir the fuel bed unnecessarily; shake the grates to clean the fire only once per day for average weather and preferably in the morning.
4. For small furnaces, pieces of coke should not be too large; size should be limited to ½ to 2 in. and uniformity in sizing is preferable.
5. Ashes should not be allowed to accumulate under the grate.

RELATIVE COSTS OF COAL AND COKE TO CONSUMER

The relative costs of coal and coke to the consumer in various districts are determined by commercial conditions, and not by the relative B.t.u. values, since these values cannot practicably be demonstrated to the consumer. In quite a number of places, domestic coke is sold to the consumer at prices about $1 per ton under the price of anthracite, since coke competes with anthracite rather than with soft coal, as a domestic fuel.

Of course this difference in price between coke and anthracite is not made because coke is a poorer fuel, because it is not. Especially since the war, the ash in coke would probably average from 8 to 10 per cent lower than the ash in anthracite. The production of dust from handling coke is no greater than from anthracite. A coke fire is more easily controlled than an anthracite fire, when one learns how, and the only disadvantage of coke is that in a heating furnace built for anthracite, the fire pot is too shallow for obtaining the best possible result with coke. In operating a heating furnace built for anthracite, the use of coke would probably call for attention once more per day than with anthracite, in cold winter weather. As soon as heating furnaces are built with fire pit designed for coke, however, the latter can be demonstrated to be a better fuel to the domestic consumer on practically every count.

CONCLUSIONS

Summing up, it appears that in the past the people of this country have been negligent and improvident. The wild animal life of the country has been sacrificed at the whim of the pioneers. The native fertility of the soil has been largely exhausted; the timber supply and the natural gas and the oil fields are pretty well exhausted.

The by-product coke oven industry promises a method of utilizing our great resource of coal in such a way as to make the loss of our other resources less embarrassing. The productivity of the soil may be maintained by the ammonia recovered from coal. Benzol may be used to replace the gasoline now obtained from the oil fields; coal tar may be used with paper as a substitute for shingles; and creosote oil may be used as wood preservative, thus prolonging the life of our timber supply. Natural gas may be replaced by coke-oven gas. The main thing for us to do is to use intelligently the resources that are left to us.

The houses in the cities can be equipped with furnaces designed and built to give better results from coke than can be obtained from any other solid fuel, and this step will aid materially in the conservation of our fuel supply. It seems to me that this Society should be the logical organization to carry out such a program. Would it not be worth while for you to give this question of furnace design your earnest consideration?

DISCUSSION

M. L. Foote: If we should redesign our boilers in order to burn coke, which seems quite essential, how are we to obtain the coke? The question for the past 25 years is not one of burning coke, as we all have recognized the importance and the advisability. How are we to get it? What system have we in Cleveland, with thousands of boilers and thousands of users wanting coke, crying for coke, and we can't get it? If the Solvay system will produce a plant here, we will redesign the boilers and we will accomplish something. On this question the Semet-Solvay Company and the consulting engineers should work together.

Author: The Semet-Solvay Company has a large plant in Detroit and dealers in this locality who care to handle coke, can get all the coke they want from there.

In Chicago, we also have a plant and Solvay coke has been advertised quite extensively. The same is true of Milwaukee. In St. Louis, Indianapolis and Terre Haute, I believe there are plants specializing in domestic coke. In the east, in Boston, Newark, N. J., Camden, N. J., and Chester, Pa., there are companies producing domestic coke. During the war they made a great mistake. Due to the enormous demand for metallurgical fuel they neglected the domestic coke business because they could get a higher price for the coke going to metallurgical interests and the result was that the demand for domestic fuel was not taken care of.

J. A. Donnelly: I think that perhaps some decided progress might be made by the Bureau of Mines taking up the question of using coke as a domestic fuel in mild and up to fairly severe weather, and then mixing with it very small quantities of coal for more severe weather. It might be necessary, with our present boilers, to use coal almost entirely in the very severe weather. In that way it might be possible to introduce domestic coke with the present fire boxes.

H. M. Nobis: When I give estimates on heating work people frequently ask me what kind of fuel they should use. I always say, "If you can get coke use it. That is the best thing to have." They all raise the question of price. Now I know that coke is far more valuable than soft coal, but there arises the difference in price between the price of soft coal and coke. Last year and two years ago we paid $18 and $19 a ton for coke, which is rather high, right here in Cleveland, and you could buy soft coal then for about $11 or $12, but it is usually thought that between $4 and $5 extra for coke would not be too much.

L. C. SOULE: I move that this Society go on record as advocating the treating of all bituminous coal for heating purposes in by-product coke ovens for the purpose of not only recovering the by-products but also of transforming the fuel into coke for its advantages as a smokeless fuel.

The motion was seconded and carried.

No. 603

FRACTIONAL DISTRIBUTION IN TWO PIPE GRAVITY STEAM-HEATING SYSTEMS

ALPHONSE A. ADLER, NEW YORK, N. Y. (Member)
and
JAMES A. DONNELLY, NEW YORK, N. Y. (Member)

THE four distinct fields of service for heating by steam are buildings, vessels, railroad trains and process work. They differ sufficiently in their design and operating characteristics to warrant separate analysis.

The field of heating buildings alone is very large. They may be heated by high-pressure steam, but far more heating is done by steam at or near atmospheric pressure. There are a large variety of buildings, including small and large private houses, apartment houses, hotels, schools, churches, hospitals, theatres, office buildings, industrial buildings and groups of industrial buildings or houses in central-station heating where heat is distributed by underground mains over large areas.

The broad field of low-pressure heating may be further subdivided according to the method of steam generation and the character of the steam supply as to the allowable pressure at which steam may be used. Thus steam may be supplied at high pressure and reduced through a reducing valve to a lower pressure; it may be necessarily used at boiler pressure; it may come from the exhaust of simple or compound engines or from turbines; or it may come from the bleeder of turbines or, in the case of compound engines, from the receiver.

The present practice in large buildings or in groups of buildings where steam is supplied from a central point at low pressure, and the returns are brought back to this point, is to use some form of vacuum-return line system. This is generally true, irrespective of the character or source of steam supply.

In buildings of comparatively small size, where constant skilled attention is commercially inadvisable, and where mechanical appliances for insuring circulation are unwarranted, simpler devices for gravity return are in general use. This paper proposes to describe apparatus designed to be applicable in this field.

APPARATUS PREVIOUSLY USED

The usual practice in low-pressure steam plants for heating buildings of comparatively small size is to attach a pressure regulator of some form

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to the boiler or source of supply. The types used are diaphragm valves, pilot-operated diaphragm valves or float valves in the return main at the boiler. In the diaphragm valves, increase in sensitiveness is obtained by increasing the size of the diaphragm. For reasons of economy the diaphragm is reduced so that pressure control is obtainable only between rather wide limits. The pilot-operated diaphragm valves are an improvement so far as sensitiveness is concerned, but only applicable where there is a considerable drop in pressure. The most sensitive pressure regulators are of the float type. Even the latter is insufficient for the small changes of pressure required for fractional control at the boiler for all outside temperatures. The object of the above apparatus was to maintain a constant pressure and hand valves were used to regulate the steam supply to the individual radiators.

Thermostatic control has had for its object the regulation of the steam supply in proportion to the outside temperature. It has taken two forms; one governing the temperature of each room independent of other rooms, and the other governing all rooms from a typical point.

GENERAL PROBLEM TO BE SOLVED

The problem here attempted is two fold:

1. To control the rate of steam flow, at the chosen point of distribution, in proportion to the outside temperature.
2. To distribute the steam to each radiator in proportion to its requirements for any and all loads.

For accurate distribution, effective methods must be used to insure dry steam by suitably dripping mains and by the cleaning of boilers. Moreover, the drop in pressure through mains, fittings and orifices of inlet valves must be computed with sufficient accuracy to assure fractional distribution to all radiators at any and all loads. To this end, it may be desirable to review briefly the factors that control the solution of this problem.

HEAT LOSSES FROM BUILDINGS FOR VARYING CONDITIONS

Heat losses heretofore have been determined only for maximum requirements which occur at times of minimum outside temperature. For convenience in making comparisons between the results of different investigators, the results have been reduced to the heat transmission per degree per square foot per hour basis. Such results are tabulated in the text-books on heating.

The conditions influencing the heat losses from buildings depend upon the following:

1. Loss due to radiation (nature and color of outside surface).
2. Loss due to conduction (material and form of construction).
3. Loss due to convection (both as affected by wind and by change in density of contacting air).
4. Loss due to infiltration (particularly at high winds).

5. Reduction of heat loss due to blinds, shades, storm doors, etc.
6. Gain of heat by building due to sunshine.
7. Gain of heat due to lighting and cooking apparatus, occupancy or other sources.

While insufficient data are available to fix either the magnitudes of each of the above factors, or their proportionate amounts for less than maximum conditions, sufficient data have been published[1] to obtain satisfactory results in most cases.

STEAM REQUIREMENTS FOR VARYING CONDITIONS

It has been customary in the past to assume that the heat losses are in direct proportion to the difference between the inside and outside temperatures. On this assumption, the steam requirements are in direct proportion to the temperature differences if the gain of heat due to cooling the condensation be ignored.

In house heating, where the water of condensation is returned to the boiler, the greatest thermal economy is obtained when the water of condensation is returned to the boiler at the lowest possible temperature. However, the heat should be liberated where it is most effective and not by cooling through the basement returns, when it is not required. In this case, the covering of returns is desirable.

In street steam supply, whether the water of condensation is trapped to the sewer or returned to the central station, it is to the consumer's advantage to abstract as much heat as possible from the condensate. To this end it has been the practice to add about 20 per cent more radiation than that needed for the maximum heating requirements in order to cool the water to as near room temperature as possible. In gravity return apparatus, about 10 per cent excess radiation is installed to accomplish the same result.

CONTROL OF THE STEAM AT THE BOILER

The control of the rate of steam generation is accomplished by the regulator shown in Fig. 1, attached to a boiler. The water of condensation flows through the pipe A and drops into the cup B in the body of the regulator. This cup is provided, near the bottom, with an orifice C, through which the water passes into the body of the regulator D, from which it is conducted to the boiler through the pipe E. The cup is supported on an arm which is pivoted at the point F, the spindle of which passes through a stuffing box to the outside of the tank, where it is provided with the lever G, which is attached by chains to the dampers in the usual manner. As the water drops into the cup, it will build up a certain head. This head will vary in proportion to the amount of water entering—the greater the rate at which the water enters, the higher the water level; the less the rate the lower the level. The weight of this water is balanced by means of a movable weight K, set at some point on the lever G, which will control the draft for any desired rate of burning. If the water

[1] The Establishment of Standard Methods of Proportioning Direct Radiation, by J. A. Donnelly, Vol. 21, Transactions (1915), p. 528.

returns to the cup faster than necessary to maintain this head, the cup will descend, raise the weight K, and close the draft I. The check draft L opens only when the air supply through the damper I is completely shut off, which is affected by proper construction of the dampers, as shown in the detail. When the damper shuts off, the amount of condensation returning to the cup will decrease, the water will eventually return to its original level, again opening the draft to the proper position for the original rate of burning.

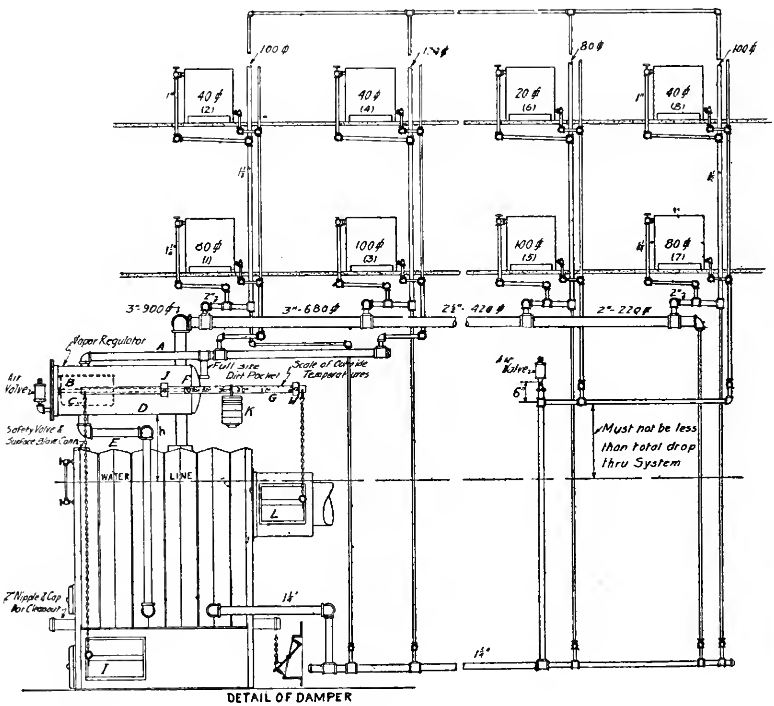

FIG. 1. GENERAL ARRANGEMENT OF VAPOR REGULATOR SHOWING APPLICATION TO A STEAM BOILER

The scale lever is so marked as to make possible the maintenance of a constant room temperature for all outside temperatures. The readings are such that if the outdoor temperature is 40 deg. fahr. and the weight K is placed on the 40 deg. division, the chosen room temperature, usually 70 deg., will be maintained. The scale on the lever may, however, be graduated for any other room temperature desired.

As shown on the drawing, it is not necessary to pass all of the returns through the regulator. It is sufficient to choose a typical section fairly representative of the entire system. The drips from the steam mains should not be returned through the regulator, as they frequently contain water due to boiler priming and never represent condensation from the radiators.

Where gas or oil is used as a fuel the regulator controls the valve supplying the fuel and the arrangement of the chains is made to suit the new condition.

Where steam is furnished from outside sources as in district steam-heating, the damper control is replaced by a butterfly valve in the steam main. The general operation of the regulator is then substantially the same as for a coal-fired boiler.

SETTING WEIGHTS ON THE REGULATOR

When the apparatus is cold, remove the weight K, place the weight H at a fixed point near the outer end of the lever and adjust the weight J at the other end so that the apparatus is balanced. Start up the fire, place the weight K at the outer mark on the lever which indicates zero outside temperature, and after the desired radiation for zero weather has become heated, remove some of the weight from K until the regulator is again in balance. Thereafter, the apparatus will not require further adjustment for maximum condition.

DISTRIBUTION SYSTEM

To effect proper distribution of the steam to the various radiators, the fractional distribution valve is used as shown in Fig. 2. It consists of an orifice in each half of the union of the form shown in Fig. 3. By rotating one-half of the union relative to the other, any opening between the minimum and maximum is possible. The area required for any size or location of radiator will be subsequently shown.

The general plan of operation will have a uniform total pressure drop to each radiator through mains, fittings, valves and orifices. The adjustment of the orifice regulates the total pressure drop to that required for uniform distribution.

PRESSURE DROP IN PIPE

Chart 1 shows the relation between radiation or total weight of steam delivered per hour and the pressure drop per 100 ft. for various sizes of pipe. The equation upon which this curve is based is that of Unwin:

$$W = 87\sqrt{\frac{PDd^5}{l\left(1+\frac{3.6}{d}\right)}} \qquad (1)$$

where W = lb. of steam delivered per min.

P = pressure drop in lb.

D = density of steam in lb. per cu. ft.

d = diameter of pipe in in.

l = length of pipe in ft. (taken as 100 in this paper).

The density D is practically constant in low-pressure steam distribution, here taken as 0.0373 for atmospheric pressure; l is assumed constant at 100 ft., and for a particular size of pipe d is constant; hence the equation may be written:

$$W_1 = k_1 \sqrt{P_1} \tag{2}$$

where k_1 is a constant the magnitude of which is equal to

$$k_1 = 87 \sqrt{\frac{0.0373\, d^5}{100(1 + \frac{3.6}{d})}} \tag{3}$$

The curve is computed on the assumption that each square foot of radiation and its connected piping condenses 0.3 lb. of steam per hr.,[1] under standard conditions of radiator 210 deg., room 70 deg. For conditions other than standard, the reading in radiation supplied is replaced by weight of steam delivered in lb. per hr.

PRESSURE DROP IN FITTINGS

Since valves and fittings may have their resistances expressed in resistance of equivalent length of pipe, the curve for pressure drop in pipe may also be used for the friction of valves and fittings. The method used in a paper by D. C. Foster,[2] who has used Meier's[3] equations for the comparative resistance of pipes and fittings of various kinds for high-pressure steam, is applicable here. The corresponding equations for low pressure are:

$$P_f = \frac{0.0257\, W\, v^{1.97}\, L_e}{288\, g\, d^{1.16}} \quad \ldots.(4) \qquad P_r = \frac{1.07\, W\, r\, v^{1.97}}{288\, g} \quad \ldots.(5)$$

where P_f = pressure drop due to friction of pipe in lb. per sq. ft.
P_r = pressure drop due to resistance in fittings in lb. per sq. ft.
W = weight of steam in lb. per hr.
v = velocity in ft. per sec.
L_e = equivalent length of pipe in ft.
g = acceleration of gravity (32.2 ft. per sec. per sec.).
d = diameter of pipe in ft.
r = factor of resistance experimentally determined.

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[1] Report of Research Committee, *National District Heating Association,* Proceedings, 1920, p. 67.

[2] Effect of Fittings on Flow of Fluids through Pipe Line, by D. E. Foster, *Mechanical Engineering,* November, 1920, p. 616.

[3] Mechanics of Heating and Ventilating, by Konrad Meier, McGraw-Hill, 1912.

To find the equivalent length of pipe, i. e., a length of pipe such that the drop of pressure in the pipe is equal to that in the fitting, equate (4) and (5); hence

$$\frac{0.0257\, W\, v^{1.97}\, L_e}{288\, g\, d^{1.16}} = \frac{1.07\, W\, r\, v^{1.97}}{288\, g} \tag{6}$$

from which by reduction and transposition $L_e = 41.7\, r\, d^{1.16}$ (7)

Table 1 gives values for various fittings expressed in equivalent length of pipe in ft. as computed from Equation 7.

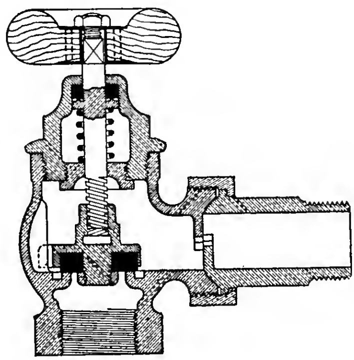

FIG. 2. CROSS-SECTIONAL VIEW OF FRACTIONAL DISTRIBUTION VALVE

PRESSURE DROP THROUGH ORIFICES

The general law of flow through orifices as given in numerous texts on the subject is:

$$Q = c\, A\, v \tag{8}$$

where Q = quantity delivered in cu. ft. per sec.
c = an experimentally determined constant here assumed at 0.62 and which remains constant for a wide range of forms.
A = area in sq. ft.
v = velocity in ft. per sec.

From mechanics: $v = \sqrt{2\, g\, h}$ (9)

where h = hydrostatic head in ft.

But since the head is proportional to the pressure by the relation

$$P = h\, D \tag{10}$$

where P = pressure in lb. per sq. ft.; D = density in lb. per cu. ft., substituting the value of h in (10), in (9) there results

$$v = \sqrt{2\,g\,\frac{P}{D}} \tag{11}$$

Insert (11) for the value of v in (8), hence $Q = c\,A\sqrt{2\,g\,\frac{P}{D}}$

To convert the quantity Q into weight W in lb. per min. it must be multiplied by the density D times 60, or

$60\,Q\,D = 60\,D\,c\,A\sqrt{2\,g\,\frac{P}{D}}$, so that on reduction and substitution of W for $60\,Q\,D$, $W = 60\,c\,A\sqrt{2\,g\,P\,D}$ (12)
for weight W per hour.

Since again, for any area A, all factors are constant except W and P, the equation may be written in the general form, $W_2 = k_2\sqrt{P_2}$ (13)
where P_2 is pressure in lb. per sq. in. instead of per sq. ft. The constant k_2 must accordingly have the value:

$$k_2 = \frac{12 \times 60\,c\,A}{144}\sqrt{2\,g\,D}$$

TABLE 1. RESISTANCE OF VARIOUS FITTINGS IN TERMS OF EQUIVALENT LENGTH OF PIPE

Nominal pipe size, in in.	Gate Valve	Std. Elbow or on Run of Tee red. in size ¼	Angle Valve and Rad. Valve	Close Return Bend	Tee through Side Outlet	Globe Valve
Factor of Resistance, r	0.25	0.67	0.90	1.00	1.33	2.00
½	0.260	0.697	0.936	1.04	1.38	2.08
¾	0.423	1.13	1.52	1.69	2.25	3.38
1	0.585	1.57	2.11	2.34	3.12	4.68
1¼	0.740	1.98	2.66	2.96	3.94	5.92
1½	0.938	2.51	3.38	3.75	5.00	7.50
2	1.30	3.49	4.69	5.21	6.92	10.4
2½	1.69	4.53	6.08	6.76	8.99	13.5
3	2.08	5.59	7.51	8.34	11.2	16.7
3½	2.50	6.70	9.00	....	13.3	20.0
4	2.93	7.03	10.5	....	15.6	23.4
4½	3.08	7.44	11.1	....	16.4	24.6
5	3.78	10.1	13.6	....	20.1	30.2
6	4.70	12.6	16.9	....	25.0	37.6
7	5.63	15.1	20.3	....	30.0	45.0
8	6.55	17.6	23.6	....	34.8	52.4
10	8.45	22.6	30.4	....	45.0	67.6
12	10.4	28.	37.5	....	55.5	83.4

RADIATION SUPPLIED BY A GIVEN AREA OF ORIFICE

The weight of steam delivered through an orifice is from the law expressed in (8), directly proportional to the area. To obtain a reasonable capacity of inlet valves, a minimum pressure drop of 2 oz. has been assumed for the orifices used in the distribution valves. Thus, if the area A be assumed at 1 sq. in., $k_2 = 4.8$, and therefore for a pressure of 2 oz., or $P = 1/8$ lb. per sq. in., $W_2 = 4.8\sqrt{1/8} = 1.7$ lb. per min.

Since standard direct-radiation without connected piping is computed on the basis of 0.25 lb. of steam per hr. per sq. ft., the radiation supplied by the steam passing an orifice of 1 sq. in. under the given conditions is $1.7 \times 60 \times 4$, or approximately 400 sq. ft. Hence, the area of orifice required per square foot of radiation is

$\frac{1}{400} = 0.0025$ sq. in. Thus, in the application at maximum load, 70 deg. fahr., inside and radiator at 210 deg. fahr., each 0.0025 sq. in. of orifice area will supply sufficient steam for each square foot of radiation. For

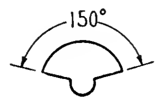

FIG. 3. FORM OF OPENING USED IN FRACTIONAL DISTRIBUTION VALVE

greater amounts of radiation, the area of orifice will be directly proportional thereto.

In Chart 2, radiation is plotted against pressure drop for various values of area of orifice. The application of the curve to practical problems will be shown later. It may be used for other amounts of radiation than those shown. For example, the scale readings of radiation might be increased ten times, in which case the areas of orifice must be increased in the same ratio, while the scale of pressure drop remains the same.

Table 2 shows the areas for various angles of adjustment of the different sizes of valves.

DISTRIBUTION OF STEAM SUPPLY AT VARIOUS LOADS

Assume

1. Density of steam substantially the same for the pressure drop used in this discussion.
2. Drips sufficient to eliminate water in piping and orifices.
3. Valves and fitting resistance replaced by equivalent resistance in pipe friction.
4. Speeds below critical velocities.
5. Pipe sizes substantially 1 oz. drop per 100 ft.

TABLE 2. ORIFICES IN SQ. IN. FOR VARIOUS SETTINGS OF UNIONS IN DISTRIBUTION VALVES

NOTE: The minimum area of opening in each size from 0 to 30 deg. is constant.

Degree of opening	½	¾	1	1¼	1½	2
30	0.0075	0.015	0.030	0.045	0.075	0.12
31	0.008	0.016	0.032	0.048	0.08	0.128
32	0.0085	0.017	0.034	0.051	0.085	0.136
33	0.009	0.018	0.036	0.054	0.09	0.144
34	0.0095	0.019	0.038	0.057	0.095	0.152
35	0.01	0.02	0.04	0.06	0.1	0.16
40	0.0125	0.025	0.05	0.075	0.125	0.20
45	0.015	0.03	0.06	0.09	0.15	0.24
50	0.0175	0.035	0.07	0.105	0.175	0.28
55	0.02	0.04	0.08	0.12	0.2	0.32
60	0.0225	0.045	0.09	0.135	0.225	0.36
65	0.025	0.05	0.1	0.15	0.25	0.40
70	0.0275	0.055	0.11	0.165	0.275	0.44
75	0.03	0.06	0.12	0.18	0.3	0.48
80	0.0325	0.065	0.13	0.195	0.325	0.52
85	0.035	0.07	0.14	0.21	0.35	0.56
90	0.0375	0.075	0.15	0.225	0.375	0.60
95	0.04	0.08	0.16	0.24	0.4	0.64
100	0.0425	0.085	0.17	0.255	0.425	0.68
105	0.045	0.09	0.18	0.27	0.45	0.72
110	0.0475	0.095	0.19	0.285	0.475	0.76
115	0.05	0.1	0.2	0.3	0.5	0.8
120	0.0525	0.105	0.21	0.315	0.525	0.84
125	0.055	0.11	0.22	0.33	0.55	0.88
130	0.0575	0.115	0.23	0.345	0.575	0.92
135	0.06	0.12	0.24	0.36	0.6	0.96
140	0.0625	0.125	0.25	0.375	0.625	1.00
145	0.065	0.13	0.26	0.39	0.65	1.04
150	0.0675	0.135	0.27	0.405	0.675	1.08
155	0.07	0.14	0.28	0.42	0.7	1.12
160	0.0725	0.145	0.29	0.435	0.725	1.16
165	0.075	0.15	0.3	0.45	0.75	1.2
170	0.0775	0.155	0.31	0.465	0.775	1.24
175	0.08	0.16	0.32	0.48	0.8	1.28
180	0.0825	0.165	0.33	0.495	0.825	1.32
Increment per deg.	0.0005	0.001	0.002	0.003	0.005	0.008

Consider two radiators and their connected piping such that the orifices are properly adjusted to give the required steam supply as shown in Fig. 4. Let W' and W'' be the weights of steam required for maximum condition and let the pressure drops in pipes and orifices be represented by P_1, P_2, etc.

For correct distribution the total pressure drops to both W' and W'' should be equal.

Thus for maximum loads $P_1 + P_3 = P_1 + P_2 + P_4$

$$\tag{14}$$

So that correct adjustment of orifices is obtained when

$$P_3 = P_2 + P_4 \tag{15}$$

or the orifice adjustment must be such that the pressure drop will compensate for the additional pressure drop in the piping connected to the farthest radiator.

It was shown (Equation 2) that the law of flow for the piping alone may be put in the form: $W_1 = k_1\sqrt{P_1}$; and for the orifices the law was shown in Equation 12 to be: $W_2 = k_2\sqrt{P_2}$.

Thus the form of the equations for both are in the same general form so that each radiator obtains its supply governed by the general law which might be written: $W = k\sqrt{P}$ (16)

Assume now that only a part of the radiation is to be heated, say $1/n$.

The general law for the new condition may be written

$$\frac{W}{n} = k\sqrt{P_n} \qquad (17)$$

where P_n is the new pressure drop since the elements that make up the constant k remain unchanged. Hence the pressure drops in the pipes and orifice are as follows:

For pipe 1 $\frac{W' + W''}{n} = k_1\sqrt{P_{n1}}$; For pipe 2 $\frac{W''}{n} = k_2\sqrt{P_{n2}}$;

For orifice 3 $\frac{W'}{n} = k_3\sqrt{P_{n3}}$ For orifice 4 $\frac{W''}{n} = k_4\sqrt{P_{n4}}$;

Since again, for correct distribution: $P_{n1} + P_{n3} = P_{n1} + P_{n2} + P_{n4}$ or $P_{n3} = P_{n2} + P_{n4}$ hence from above

$$P_{n3} = \left(\frac{W'}{n\,k_3}\right)^2; \qquad P_{n2} = \left(\frac{W''}{n\,k_2}\right)^2; \qquad P_{n4} = \left(\frac{W''}{n\,k_4}\right)^2;$$

$$\text{or } \left(\frac{W'}{n\,k_3}\right)^2 = \left(\frac{W''}{n\,k_2}\right)^2 + \left(\frac{W''}{n\,k_4}\right)^2$$

Cancelling the n which appears as a factor in all terms we find that we have,

$$\left(\frac{W'}{k_3}\right)^2 = \left(\frac{W''}{k_2}\right)^2 + \left(\frac{W}{k_4}\right)^2 \qquad (18)$$

Thus the pressure drops for $\frac{1}{n}$th part of the load is independent of n and therefore such that the proportionate distribution is the same as that at full load.

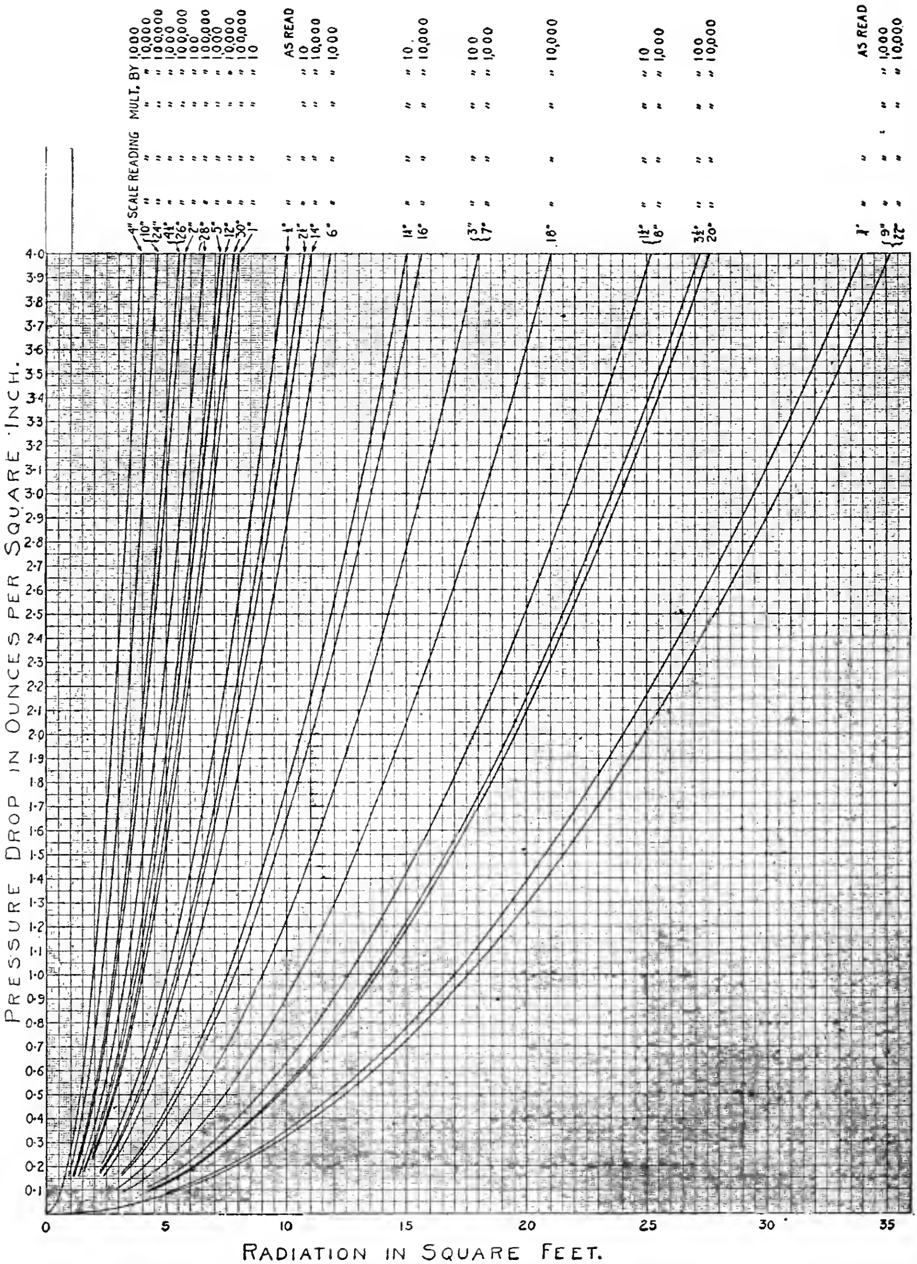

CHART 1. CURVES SHOWING RELATION BETWEEN RADIATOR AND PRESSURE DROP PER 100 FT. FOR VARIOUS SIZES OF PIPE; RADIATORS AT 210 DEG. FAHR. AND ROOM TEMP. AT 70 DEG. FAHR.

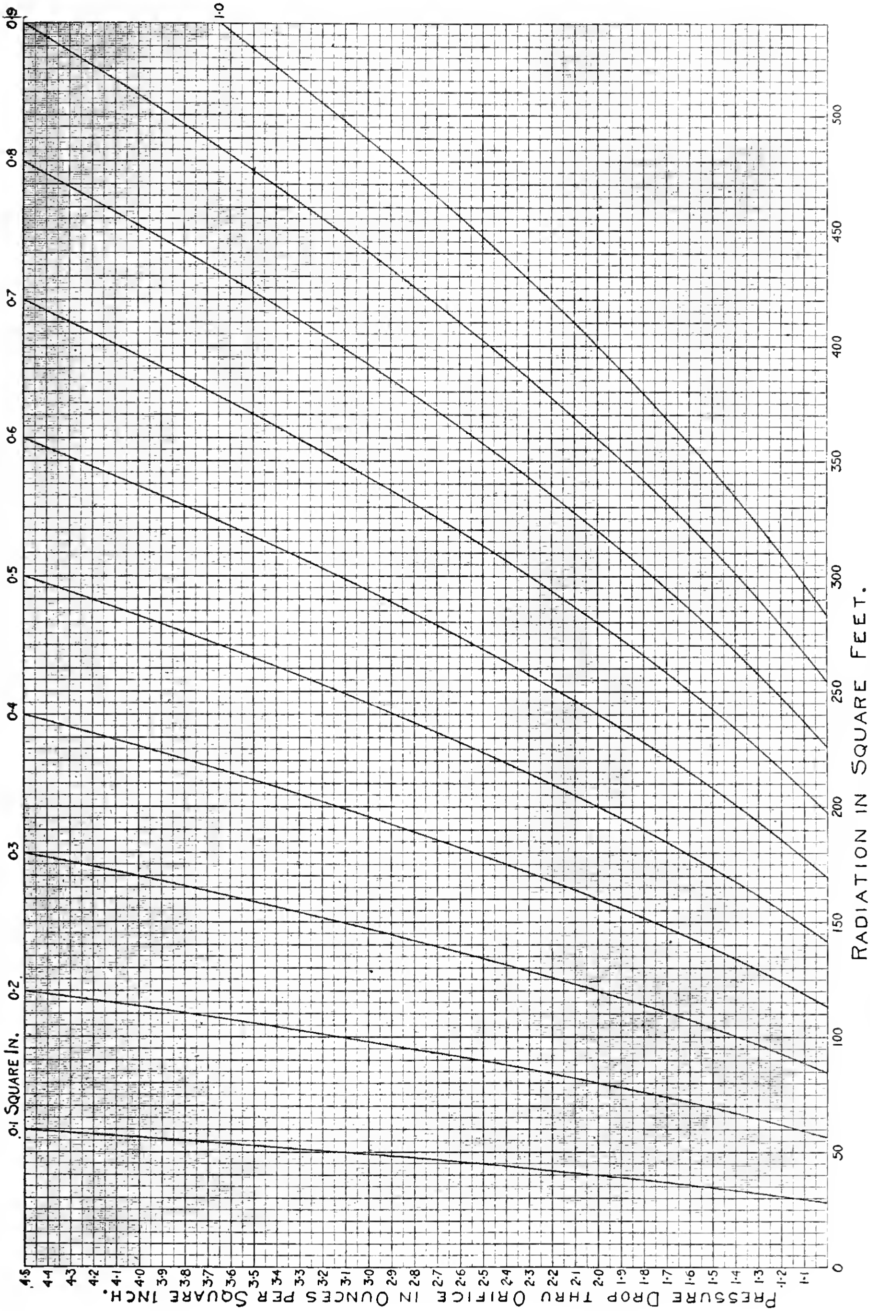

CHART 2. CURVES SHOWING RELATION BETWEEN RADIATION AND PRESSURE DROP THROUGH ORIFICE FOR VARIOUS AREAS OF OPENING

PRACTICAL APPLICATION OF CURVES AND TABLES

Assume the amount of radiation to be supplied is as shown in Fig. 1. The pipe sizes are chosen on the basis of 1 oz. drop per 100 ft. of pipe and may be taken from Chart 1. These are indicated in Fig. 1.

In general, assume that the orifice in the valve of the radiator farthest from the boiler has a drop of about 2 oz. Due to the radiation beyond radiator (8), assume that the orifice drop for (8) becomes 2½ oz. This then becomes the starting point in this calculation. The total pressure drop from the boiler to radiator (8) when computed is, for example, 1.25 oz., for pipe and fittings. This added to the drop through orifice of 2½ oz. gives a total drop of 3.75 oz.

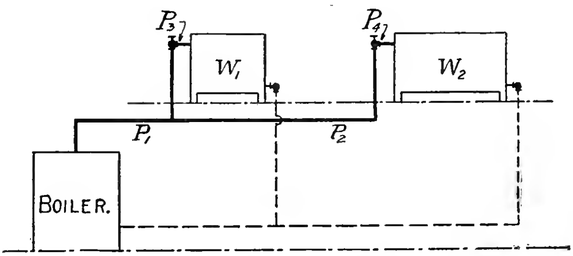

FIG. 4. DIAGRAMMATIC ARRANGEMENT OF RADIATORS AND PIPING USED IN ANALYSIS OF CORRECT DISTRIBUTION

From Chart 2, for 40 sq. ft. of radiation at 2½ oz. drop, it will be found that 0.09 sq. in. of area will be required. From Table 2 this corresponds to a 1 in. valve, set at 60 deg. of opening. Suppose that the total drop from the boiler to the orifice for radiator (7) through pipe fittings and valves is equal to 1.051 oz. This leaves the drop to be taken up by the orifice on radiator (7) of 3.75—1.051 or 2.699 oz. From Chart 2, this corresponds to an area of 0.173 and from Table 2, to an angle of 73 deg. on a 1¼ in. valve. Thus the plan is to make the total drop through pipe from boiler to radiator (7) equal to the total drop through pipe from boiler to radiator (8), i. e., 3.75 oz.

If the length of run to radiators (2) and (1) is assumed to be 100 ft. less than to radiators (7) and (8), the drop in pressure to these radiators will be about 1 oz. less; therefore the orifices on (2) and (1) must be adjusted for a drop of 1 oz. greater than in the former case.

If economy in pipe sizes is desired, the nearby radiators might be piped in somewhat smaller sizes, thus increasing the drop through the pipe and correspondingly decreasing the drop through the orifices.

APPROXIMATE RULE FOR DETERMINING ORIFICES

With conservative sizing of pipe and with some regard for critical velocities, the rough rule of 1 oz. drop per 100 ft. of run gives a drop of

0.1 oz. per 100 ft spacing of radiators. Hence, if the radiator farthest from the boiler has a 2 oz. drop through the orifice, each radiator every 10 ft. nearer the boiler must have approximately a 0.1 oz. greater drop through the orifice.

In case of error in setting the orifices, it is possible to estimate the percentage of overheating or underheating. In this case, decrease or increase respectively, the area of the orifice in direct proportion to the percentage of error.

EFFECT OF VARIABLE CONDITIONS OF OPERATION

Closing Valves on Some Radiators. Consider a house in which several rooms are normally unoccupied and in which it is desirable to have the radiators shut off. There are two ways in which this may be accomplished. The simpler of the two consists of having two weights for K shown in Fig. 1. The apparatus will be adjusted for normal operation as has been described when the normal radiation forms the heating load. If the additional radiation represents an increase of perhaps 25 per cent, then the weight K must be increased in the same ratio. If the apparatus is adjusted for the maximum load, the weight K must be decreased in proportion to the radiation shut off. The other method consists of moving the weight K to compensate for the new load. Thus if the temperature difference between inside and outside is 40 deg. fahr., and it is desirable to shut off 25 per cent of the radiation, since the heat supplied is in proportion to the temperature difference, the weight will be moved to a position corresponding to three-fourths of the temperature difference, or to make the scale reading 10 deg. fahr., less, in this case.

Opening of Windows in Rooms. Without a distribution system such as that described, a particularly annoying case is that where the occupant of a room close to the boiler opens a window on a cold day. Under these conditions, the condensation in such radiator is above normal and with a limited steam supply other radiators will fail to receive their due proportion of steam as a result of this short circuiting. With orifice control such as that proposed, the pressure between the two sides of the orifice, determines the quantity of steam passed. Thus there is a predetermined limit of the supply to this radiator and the effect is that of reducing the temperature only in such rooms where the open windows exist. The steam supply to other radiators will not be effected, although the temperatures of the rooms in which they are located may be slightly reduced, due to the increased transfer of heat to rooms where the temperature is lowered.

Additions and Changes to Plant. A well laid-out distribution system which insures proper supply to all radiators for the original installation, is frequently disturbed by the addition or relocation of radiators. Where the pressure drop is solely that due to friction in valves and piping, no simple remedy is available. If a certain amount of this pressure drop is available by change in orifices, poor distribution may be corrected in the new condition.

Irregularities in Daily Performance. Occasionally erratic operation of certain plants is observable through the failure of certain radiators to heat from day to day. The same radiators are not always affected, for the trouble seems to wander to different parts of the heating system. This condition is always due to mixtures of steam and water in the mains, risers or radiator connections. The cause may be poor design of boiler or piping, insufficient drips, trapped radiator connections, dirty water in the boiler, or a "critical velocity" in the piping, so that the down-flow of water in risers and radiator connections is balanced by the up-flow of steam and thus held in suspension. Some of the remedies are immediately apparent.

Connecting the tops of the risers as shown in Fig. 1 will overcome a considerable part of such trouble by equalizing the pressure. An arrangement of piping such as this will give all the advantages of a down-feed system, without the disadvantages and expense of an up-feed riser and a large supply main on the ceiling of the top floor. Any tendency of water to be held in a certain riser, thus reducing the steam pressure above, is immediately offset by a flow of steam from the other risers.

DISCUSSION

H. M. HART: I would like a little further explanation. Do I understand that when you use this condensation regulator that you do not use the steam flow control on the main from the boiler? Further do you refer to a two-pipe system?

AUTHOR (A. A. Adler): The condensation regulator is used on both one- and two-pipe systems. The steam flow is not controlled in the main but by means of an adjustable orifice at each radiator.

H. M. HART: I understood you had two methods. One was to control the volume of steam supplied into the main before it went into the system; the other was the condensation.

AUTHOR (A. A. Adler): No, the condensation controls the rate of burning and this in turn determines the rate of steam flow.

H. M. HART: I want to get more light on the condensation control. When you would start up in the morning the condensation would be very rapid. Wouldn't there be a tendency to choke down the fire to such an extent it would take a long time to fill the system?

AUTHOR (A. A. Adler): Yes, there would be a tendency to choke if the weight on the lever is set to maintain the outside temperature. The remedy would be to adjust for a higher rate of burning until the rooms are brought up to temperature and then readjust to maintain the required temperature.

H. M. HART: I agree with you that if the system was full of steam this regulator would function properly; but I do not believe the rate of condensation would be the same or uniform. When you are starting the system the rate of condensation would be a great deal more rapid than it would be if the system was operating continuously, because you are putting steam into a cold system, cold radiator; and, in fact, that is the peak load on your boiler. Your regulator would be set for the normal rate and your boiler would be called on for the peak load. As a matter of fact, you have the peak load on your boiler nearly every morning, unless you carry steam all night.

Author (A. A. Adler): What we recommend is that at night, instead of setting the weight at the outside temperature, you set for a 20 deg. difference of the thermometer. That is, if your outside temperature is 30 deg. you set it for 40 or 50 deg., thereby cutting down the temperature of the room in the direct proportion; that is, having the room at 50 or 60 instead of 70 deg.

In the morning the condition occurs that Mr. Hart speaks of, and the usual practice is to check down the fire thoroughly, fill the firepot full of coal, open the ash-pit door and let it stay open till you clean the ashes out; then let it remain open until considerable excess steam is made and a much larger rate of burning is attained. In that way pressure is kept in excess of that required for the outside temperature, until the building is brought up to temperature. Then the ash-pit door can be closed and the weight adjusted to the normal rate of burning.

H. M. Hart: I want to get some further information. Do I understand that you eliminate the control at the radiator after it is once set? In other words, don't you have to have a control valve that you can shut off at the radiator if you want?

Author (J. A. Donnelly): Yes, there is a control valve to shut off the radiator, but it is intended to leave the radiator valve alone and do all the regulating at the boiler unless you want one room colder than another.

H. M. Hart: I understand this to be a closed system. After you drive the air out, then a vacuum is maintained. It has been my observation that where you have a closed system it is very difficult to maintain uniform conditions on the flow of steam into the radiators, because as the radiator starts to cool then a condition of vacuum is created which unbalances that radiator from the rest of the system. I have come to the conclusion that the only way you can get a uniform control is by having a uniform pressure and the only way to obtain that is to eliminate the vacuum feature and have an open system, so that as the radiator cools it will draw air back into the radiator.

Author (J. A. Donnelly): I thoroughly agree with you, we do not recommend it, although it might be used—some people want to use vacuum but we do not usually recommend the vacuum feature on the two-pipe system. We do let the air come back. After the radiator is heated to excess in the morning in order to bring the building up to temperature, the pressure is dropped to that required for normal operation. We did not want to add too many curves to this paper, but if the total system will circulate at 4 oz. you can calculate the pressure which must exist in the boiler for all outside temperatures, given say two-thirds or three-fourths the amount of steam, because we can figure under those conditions the comparative drop in the steam main. I may say that the steam which would circulate at the 4 oz. end in zero weather would need only 2 oz. at approximately 20 deg. above zero to give the proper amount of steam.

H. M. NOBIS: Referring to what Mr. Hart said regarding the amount of condensation in the morning, we made some tests in our shop to find out how much condensation is drawn over the first hour by the heating system, and we found that 33 1/3 per cent of the total condensation returned to the dry return pipe during that period, was carried over by the steam connection of the boiler. During the second hour 9 per cent was carried over and during the third hour 6 per cent.

Regarding introducing an orifice in the piping system, we have always regarded friction as an arch enemy of the system and it has been the practice to eliminate it as much as possible.

AUTHOR (A. A. Adler): I do not know whether the last speaker is correct in saying that friction is detrimental.

Friction itself, from a thermo-dynamic standpoint, as applied to a heating system, is not a loss. The object is to give up the heat to the building and it is necessary to have the piping and radiators lose heat. The important thing to do is to control friction. In practice it is desired to make the most economical pipe connections, and leave the inherent pipe friction alone but absorb the remaining pressure drop by some form of orifice in order to get control of the radiator. This is the function of the control system as described in the paper.

J. C. HOBBS: It seems to me that this paper is an example of good practical engineering. The authors are to be commended for the thorough presentation of their subject. Instead of using a lot of special pipe sizes in order to get equal distribution in the various parts of the heating system, the simple expedient is used of introducing an adjustable friction into the pipes serving those radiators which on account of their location are given an excess supply of steam.

It is not my intention to advocate the introduction of more friction into any piping system, but inasmuch as it is impossible to eliminate all friction and a certain amount of friction must be provided for in connection with those radiators located furthest from the source of supply, then it seems logical to equalize the action of the radiators by introducing a sufficient amount of friction in connection with those radiators located close to the source of supply, so as to insure equal distribution. This idea can be carried still further if it is desired and certain radiators which are in reality too large, made to operate at lower capacities by reducing the capacity of the steam supply line.

As has been pointed out, I believe one of the merits of this plan is that the distribution is practically correct for all loads if it is once set correct for any load. The friction characteristics of straight pipe and of special resistance are the same, namely, that the friction head increases approximately as the square of the velocity.

A few minutes ago one of the speakers spoke of boilers carrying water out with the steam when they were being forced. Such a condition is very liable to happen in connection with low-pressure boilers when operated at high ratings, on account of the enormous volume of steam at low pressure. I believe few engineers realize that a boiler when operated at 5 lb. pressure, delivers seven times the volume of steam that it would, if operated at 150 lb. pressure per sq. in., even though the actual amount of heat discharged is exactly the same. This means one thing, that is, that the velocities of the steam leaving the boiler are seven times as high, and that the chances for water to separate out from the steam are very much less at low pressure than at high pressure. More liberation space should be provided and special attention given to dry pipes in those cases where low pressure boilers are operated above the normal rate.

Even at 75 lb. per sq. in. gauge, a 600 hp. boiler with which I am familiar, when operated at 400 per cent of rating threw considerable water with the standard dry-pipe installation. Changes in the dry pipe and the method of collecting the steam within the boiler drum have resulted in a great improvement.

Boiler ratings as high as 1000 per cent have been obtained with high-pressure boilers with perfect success, but the pressures carried were about 300 lb. per sq. in. and a special steam collecting arrangement was used.

Only one question occurs to me and that is, how can the flow control system be applied to large office buildings? It has been my experience that the actual steam demand of the building does not vary exactly in proportion to the outside temperature, but that the question of the time of day also has a great influence. I have in mind, the condition in which the occupants of the building have just arrived on a cold morning. The offices may be at a temperature of 65 or 68 deg., but on account of the occupants having been chilled during their trip to the office, they do not feel that the room is sufficiently warm and more steam is required. This same thing happens after the lunch hour.

Author (J. A. Donnelly): Where it is desirable to overheat the building in the early morning hours when the tenants are chilled the object is attained by installing extra radiation if necessary. The regulator may be put out of service for the time being by moving the weight on the lever as far as it will go or even adding additional weights for the time in which the abnormal demand exists. After the emergency is past, reset the regulator for normal operation.

There are two details which are included in Fig. 1 on page 300. In the first place, it so happens that the most convenient way to get the typical amount of condensation back to the regulator is to run a steam line and drip it at several points. Also, to take the riser connections from the top of the steam main and drip each riser separately into a drip main. At least drip the risers into the wet return and bring the returns back separately. In this way we measure the rate of condensation in the risers only and not the steam main.

Another feature is found in connecting the steam risers together at the top. That is somewhat new and I find it is very desirable in cases where the water tends to surge in the risers. The water does not surge in all risers alike nor in all of them at once. In the cases where water is present the steam can be supplied momentarily from the top. I find this to work out very satisfactorily.

H. M. HART: May I ask one more question? Have you tried the control of the steam by means of a thermostat in place of this regulator which would be actuated by the temperature of the steam in the boiler? It seems that would be more delicate and accurate than this system.

AUTHOR (J. A. Donnelly): The inherent characteristics of this regulator permit of any degree of sensitiveness desired. The cup in the regulator can be made any diameter or any height. The increase of diameter affords considerable weight for any hydrostatic head on the orifice in the cup and so will move a stiff damper mechanism. A variation in the height of the water column will permit any degree of sensitiveness desired.

H. M. HART: Suppose that you were controlling your draft by the condensation from a few radiators instead of all. Then somebody shuts off two or three of those radiators; isn't that going to throw your regulation off? For that reason wouldn't it be better to control it from the temperature of steam in the boiler?

AUTHOR (J. A. Donnelly): Where a few radiators are shut off on the typical section which is under control of the regulator, the remedy is to readjust the position of the weight on the lever as shown in the paper. In such cases where these radiators are subject to the whims of the tenants the regulator cannot be adjusted for such uncertainties.

J. M. ROBB (written): The use of the rate of condensation return flow, for the regulation of draft, or steam admission, is one of the cleverest bits of original thought applied to steam heating that has come under my observation. For certain conditions, it is likely to prove a very valuable assistance towards accomplishing a much desired effect.

But, where the paper falls short, is at a point where in my opinion, all similar efforts have fallen down. In applying this method of control, or any similar one, consideration must be given to the fact that *bituminous fuel won't respond to damper operation for the desired control, as will anthracite or other fuel low in volatile matter.* If this limitation is not borne in mind, many of those designing and installing heating plants for bituminous fuel are likely to be misled, with no small loss. I have seen many low-pressure boilers wrecked from the failure to comprehend this condition. It is a great help, towards the desired control, with bituminous fuel, to fit the chimney with a check draft *with area equal to chimney area*, and arrange this for operation from the controlling mechanism, but it is only a help.

Steam boilers of the down-draft type, for bituminous fuel, also respond more readily to control than the up-draft type, but they are not an insurance against trouble, when in mild weather, the attendant fires in the same manner as in severe weather and many of the radiators are shut off. Under such conditions, surplus heat generated will certainly make a pressure rise in the boilers that may result in such a damming back of condensation in the return system as dangerously to lower the water in the boiler. This is particularly true of the usual cast-iron boiler construction, with its low water content.

In my opinion, there is but one satisfactory way to meet this condition. That is to use a pressure regulating valve and return trap combination, to limit the pressure on the heating mains to the desired point, to insure the return of condensation against any pressure rise in the boiler, and to store surplus heat in the boiler as excess pressure.

In the past twelve years I have designed, inspected, tested and operated hundreds of heating plants, with the above combination, all over the United States. The more extensive my experience has grown with it, the greater has my respect for it become.

In one plant that I have in mind, so designed that the boiler could be operated with the above combination, at from 5 to 10 lb. pressure, reduced to ½ lb. in the heating mains, or could be operated as a direct gravity job, with not to exceed 1 lb. pressure on the boiler and mains, the fuel economy, in carefully conducted tests under identical weather conditions, was 25 per cent in favor of the combination.

To those who doubt that surplus heat can be stored in the boiler as excess pressure, let me cite a 20,000 sq. ft. installation, in five buildings, in the mountains of Georgia, with the distant building 1000 ft. from the boiler house, where the only attendant, a $10 a week negro, (that is what he was paid in 1910), puts in his last fire at eight at night, and finds heat in the radiators of the distant building, when he comes on duty at five the next morning.

I have been told, when mentioning this, that anything will heat in Georgia. As a matter of fact, I have found more heating plants that would not heat, when heat was wanted, in the southern states in which I have worked than I have found elsewhere in the country. I have a hospital with 13,000 sq. ft. of radiation in North Dakota, where we do the same thing. Not only that, but in this hospital we heat our water with an instantaneous heater, and burn lignite for fuel. During the first winter, in order to check the water heating, the authorities arose at all hours of the night to draw quantities of hot water without ever finding a shortage.

Surplus heat can be stored as excess pressure in boilers as readily as it can be stored as hot water in tanks. But any one who attempts this is likely to get astonishing revelations as to the condensing capacity of heating plants, unless he is accustomed to the restricted orifice idea, with orifices so proportioned that resistance enough will be made to flow to radiators, so that the entire steam mains will fill with steam before any radiator gets any.

The restricted orifice idea, properly understood and applied, will convince any engineer that he needs no automatic traps nor air valves on radiators. W. S. Monroe, on page 29 of the 1902 edition of his book entitled—*Steam Heating and Ventilation*—describes the use of discs with orifices 1/16 in. in diameter, in place of the automatic air valves on the radiators of a Paul system in a large Chicago office building. I have used this application many times and have seen others use it much, with increasing wonder each time as to why any one ever considers the use of an automatic valve for service like this. I have used the restricted orifice idea also to secure the more even distribution of exhaust steam in extensive plants with such results that I think it has the same possibilities for the distribution of varying quantities of heat in accordance with weather requirements, by varying pressures, as is possible with hot water, by varying its temperature.

I wish also to comment unfavorably on the style of restricted orifice shown in Fig. 2 from an inspection standpoint, as well as on the utter futility of expecting proper attention to the requirements for this method of regulating flow from steam fitters. Even when the plans are carefully drawn, with identifying numbers for each valve, properly marked, and valves shipped out with corresponding numbered tags, it is the usual experience on a large job to find any number improperly applied. With the construction indicated, it is a physical impossibility to check the adjustment of the orifices, without disconnecting unions and moving radiators, or shifting valves out of line. This is a job of some magnitude for a dozen radiators. When it is necessary for several hundred radiators, the amount of labor and time is enormous.

In a little treatise on laundry operation, a few days ago I found this comment, "When the delicacy of technique in any given test exceeds that which can be expected of the average operator, any test that might be perfectly satisfactory when applied by the inventor, because of the high degree of skill acquired by long practice, may be expected to fail." There is a world of common sense in this statement that too many designers frequently overlook. If the principle behind it is properly comprehended, no one will ever expect results from any restricted orifice application to steam heating, where the steam fitter is expected to make any adjustments. This is the belief, backed up by long and varied practice, of those who urge the use of fixed orifices that the fitters cannot vary, and with arrangements that will permit the ready checking of the steamfitters' work.

Another point that requires comment is the size of orifice, 0.0025 sq. in., mentioned for 1 sq. ft. of standard steam radiation on page 305. Ever since Tudor, in his patent granted in 1882, stated "that an orifice ¼ in. in diameter would supply all the steam at 3 lb. pressure that 200 sq. ft. of radiation would condense," there has been a great variation in the orifice areas used by the different people who have applied his practice.

For the past ten years or more, I have been using an orifice 1/32 in. x 5/16 in., as the standard for 10 sq. ft. of steam radiation, under usual conditions, with the idea of having this orifice pass 2½ lb. of steam per hr. under 1 lb. pressure.

This orifice was used in multiple for the required radiator capacity. Its primary object was to cause enough resistance to flow, so that the entire main system would fill with steam before any radiator would take any, and so that at any fraction of 1 lb. pressure, the flow of steam to the radiators would be in proportion to the pressure. I have found no reason, in hundreds of inspections, to change this orifice size. But I have found it necessary, on certain jobs where high-pressure steam was reduced to low pressure in the mains, to limit my main pressure to ¼ lb., in order to limit flow to what my radiators would condense.

I have also found other instances where 2 lb. pressure would not permit my orifices to pass the required flow of steam. And in many such, I have had endless discussions with those not familiar with restricted orifice application, who contended that the orifices were too small. I have never failed to end such discussions by blowing the boilers clean, thus obtaining steam of the quality that would give the proper flow. The authors touch too briefly on this requirement. Failure to heed it has caused many condemnations of the restricted orifice application, otherwise properly made. In commenting on opening of windows in rooms, the authors have not elaborated as much on the advantages as they might. I have inspected overhead work in tall buildings, where reduced pressures would just fill the mains with steam, with no more heat in top floor radiators than in first floor radiators.

Whenever the possibilities of the restricted orifice properly applied are generally appreciated, it is my opinion that there will be little heating of any other kind. Let me add this warning, however, to those who contemplate it. For successful results, the piping must be considered more as a reservoir, and less as a conduit. That does not necessarily mean increased cost of piping over a *properly* designed, ordinary two-pipe job. But it does mean more attention to the maintenance of pipe sizes at the most distant portions of the piping system, than those not familiar with proper application appreciate.

In the correct application, as I see it, the pressure reducing valve area may be held down to the same proportion as the orifice area for the individual radiators, with outlet extended at reduced size, either to the most distant point, and there increased to low pressure main size, or extended to some central point and expanded into a large fitting header, from which a number of small low pressure mains run.

In this construction, it is always important to use the diaphragm type of pressure regulating valve, and control the pressure from the distant end of the low-pressure mains, not from the large header. With this sort of application, I have in many instances held piping costs below well designed two pipe costs for ordinary service.

For a return trap on this class of work, I prefer the Lytton type, because there is less water line distance required than on any other. I plug the top opening of the trap, making a single riser connection to the trap receiver, from between two check valves. In most instances, the outlet check valve should be greatly reduced in size, but the pipe sizes should be maintained the same in order to avoid a nasty back slap when the boiler pressure is released from the trap. I prefer to use a small pipe receiver above the trap to take the exhaust with drop connection to the return, so that the condensed exhaust will be returned with the condensation from the plant. The receiver can be vented through an open vent pipe or air valve as desired.

Let me close this discussion by pointing out that the restricted orifice idea has great possibilities for application to ventilating fans, so that the fan will discharge or exhaust in proper proportion to all openings connected to it. I have made a number of applications of this sort, every one of which has worked out so satisfactorily that I think neither splitter nor volume dampers are necessary.

No. 604

THE APPLICATION OF GAS TO SPACE HEATING

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Non-Member

THIS paper endeavors to present some brief and general information regarding the application of gas to the heating of residences, commercial buildings, and in general, to space heating—first, as to its general properties and the conditions governing its combustion; second, with regard to the forms and types of appliances used in burning it; and third, with regard to its relation with competing fuels.

KINDS OF GAS AND THEIR PROPERTIES

Manufactured gas, as distributed in American cities, is usually either coal gas, carburetted water gas, or oil gas; the latter is confined to the Pacific Coast, and does not so closely concern us here in the East. Coal gas is produced by the destructive distillation of bituminous coal in retorts or ovens. When it is produced as the major product, with coke as a by-product, it is usually called coal gas. When gas is a by-product, incidental to the production of coke for metallurgical purposes, we speak of it as coke-oven gas. A typical analysis of coke-oven gas in percentage is as follows: H_2, 50.0; CH_4, 36.0; C_xH_y, 4.0; N_2, 2.0; CO, 6.0; O_2, 0.05; CO_2, 1.5. Its specific gravity is from 0.40 to 0.45. Its heat units per cu. ft. vary from 600 to below 500 B.t.u., depending upon the stage of the process from which it is obtained and whether it is stripped of its light oils.

Blue water gas is made by the blowing of incandescent carbon in the form of coke, or hard coal, with steam, and consists of hydrogen and carbon monoxide. This gas has a heating value of about 300 B.t.u. per cu. ft. It then passes into the carburetter where it is enriched by an oil spray and becomes carburetted water gas. Its heating value depends upon the amount of oil added to the blue water gas. Its specific gravity is about 0.6 and a typical analysis in percentage is as follows: H_2, 40.0; CH_4, 25.0; N_2, 4.0; CO, 19.0; C_xH_y, 8.5; O_2, 0.5; CO_2, 3.

Natural gas has a heating content varying from 700 to 1100 B.t.u. per cu. ft., depending upon the district where it is found. In most fields, 900 to 1000 B.t.u. is a good figure to use. Its average specific gravity is in

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the neighborhood of 0.6 and a typical analysis would be as follows: H_2, 3; CH_4, 92; C_xH_y, 3; N_2, 2.

GENERAL PRINCIPLES RELATING TO THE COMBUSTION OF GAS

The amount of air necessary to burn a cu. ft. of gas is very nearly proportional to the heat units in the gas; this holds for all combustible gases and all mixtures that do not contain non-combustible gases. This law is very little understood by the general public, and even by some engineers, yet it governs the results that can be obtained from various gases. It should always be remembered that combustion consists of the

FIG. 1. TYPICAL UNVENTED GAS-STEAM RADIATOR

FIG. 2. A DESIGN OF VENTED GAS RADIATOR

combination of the carbon and hydrogen contained in the gas with the oxygen of the air, and that what we are really dealing with and employing, is a mixture of gas and the air necessary for its complete combustion, and that the heat content of this mixture is always practically the same as long as the gas contains no inert substance such as nitrogen. This is true whether we are dealing with natural gas at 1000, or blue water gas at 300 B.t.u. per cu. ft.

This may be made to appear more clearly by illustration. Natural gas requires 10 cu. ft. of air to burn 1 cu. ft. of gas. Therefore, the volume of the air-gas mixture to contain 1000 B.t.u. will be 11 cu. ft., and the B.t.u. per cu. ft. of mixture will be 91. Coal gas having 500 B.t.u. per cu. ft., requires 4.5 cu. ft. of air for each cu. ft. of gas; therefore, the volume of the air-gas mixture to contain 1000 B.t.u. will be 11 cu. ft., and the heat content of the mixture will be 91 B.t.u. per cu. ft. The practical result is that we can take a boiler or furnace designed for natural gas at 1000 B.t.u. and get the same rating out of it with blue water gas at 300 B.t.u., provided the proper burners are furnished and correctly adjusted. The figures above are approximate and stated in round numbers.

It is important to remember that the combustible gases that we deal with in heating, are made up of hydrogen and carbon, or combinations of these two elements, and that when these two elements are completely burned, they form carbon dioxide and water vapor, all the hydrogen combining with oxygen to form water vapor and all the carbon combining with oxygen to form carbon dioxide. Both of these products are in themselves perfectly harmless, but either may be objectionable if held in

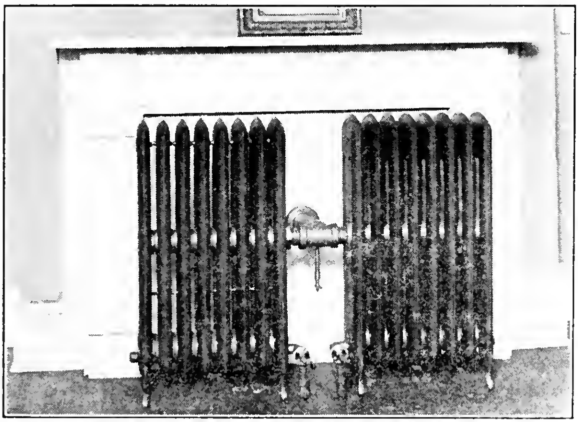

FIG. 3. AN ARRANGEMENT OF TWO VENTED GAS RADIATORS, REPLACING A MANTLE STOVE AND EXHAUSTING INTO THE CHIMNEY

a room in excess—the water vapor because it tends to condense on window panes, walls and other cold surfaces, frosting the panes and moistening the walls, and the dioxide because if it accumulates in excess, it will reduce the oxygen content of the room below the proper healthful limit.

Gas when used for heating, may be burned in two ways. In the older of these, a jet of gas issues from a narrow slot in a tip made of some refractory material and the flame spreads out in a thin sheet, expanding until it obtains enough contact with the air to secure the amount of oxygen necessary to burn the gas. This means a large low-temperature luminous flame that will deposit carbon if brought in contact with a cold surface. It therefore cannot be used in a confined space or under a boiler.

In 1855, Bunsen devised the burner, or what might more properly be termed the mixer, that bears his name, and made the first great step in the proper combustion of gas for heat production. The Bunsen principle consists of so arranging a gas orifice with respect to an air opening behind or beside it that the jet of gas on its way to the burner may draw in air from the atmosphere, which becomes mixed with the gas before reaching the zone of combustion. This pre-mixing of air and gas has the effect of giving a much shorter and more concentrated flame than the yellow flame of the non-Bunsen burner. The Bunsen flame is blue and has very little illuminating power, but its temperature is considerably higher than the yellow flame. It also has the property of burning in contact with a cold surface without depositing carbon. This fact, together with its high temperature, have caused it to be adopted for practically all burners except those used for illuminating purposes and a certain class of space heaters.

There is a popular superstition that more heat is obtained from gas when a blue flame is used. The total amount of heat developed by the combustion of a given quantity of gas is of course exactly the same whether a blue or a yellow flame is used, but the yellow is a large, low-temperature flame and the blue, a concentrated, high-temperature flame; therefore, the Bunsen burner may be more efficiently applied to most purposes and may be used in many places where the yellow flame cannot be used at all. The one advantage the yellow can claim over the blue, is that it is not subject to "flash-back."

HEATING APPLICATIONS AND APPLIANCES

Gas-heating applications may be divided into two classes. The first will contain those cases where gas is used exclusively, and the second, those where gas is used as an auxiliary to heating by coal or some other fuel. In this class, it may be used during the spring and fall, before the coal furnace is started or at the same time as the coal furnace to supplement it.

Appliances may be divided into two great divisions, the first consisting of central heating plants, that is, steam, hot water or vapor boilers, and hot air furnaces, from which heat is distributed to a number of rooms. The second contains room heaters that are placed directly in the space to be heated.

Room heaters will be considered first as they are the simplest application of gas to space heating. The form and variety of these heaters is almost infinite, but they may be classified in general in two ways; they are either vented or unvented, and they employ either yellow- or blue-flame burners.

THE UNVENTED HEATER

The unvented heater, as compared with the vented type, possesses the following advantages. It is 100 per cent efficient. It can be placed practically anywhere in the space to be heated. The installation cost is low. The variety of forms is almost infinite. They include many kinds of radiant heaters, the familiar sheet-iron heater, the gas-steam radiator and many others. They all have the disadvantage, however, that they discharge the products of combustion into the room to be heated. If combustion in the heater is perfect, these products are water vapor and carbon dioxide. Where the space is well ventilated, these products will not collect in appreciable quantities, but where the rooms are kept closed, the water will cause frosting of windows and sweating of walls and the carbon dioxide may exceed the permissible minimum.

Several years ago, the writer conducted tests in a dwelling house heated by unvented gas-steam radiators to find the probable amounts of dioxide in the air under practical working conditions. The tests were run at different times to determine the degree of vitiation of the air in the room. The first of these tests was run in a room 13 ft. 6 in. by 18 ft. by 9 ft. 2 in., with the doors closed; an eight-section gas-steam radiator was left burning for 4 hours; the following data were collected:

Time Start	Gas Consumption, cu. ft.	CO_2 per cent	O_2 per cent	CO	Room Temp. deg. fahr.	Outside Temp. deg. fahr
		0.3	20.6	....	61	35
1st hour	....24.5	0.5	20.1	....	71	36
2nd hour	....22.5	0.9	19.8	....	80	36
3rd hour	....22.0	1.2	19.4	....	83	36
4th hour	....22.0	1.6	18.9	....	86	36

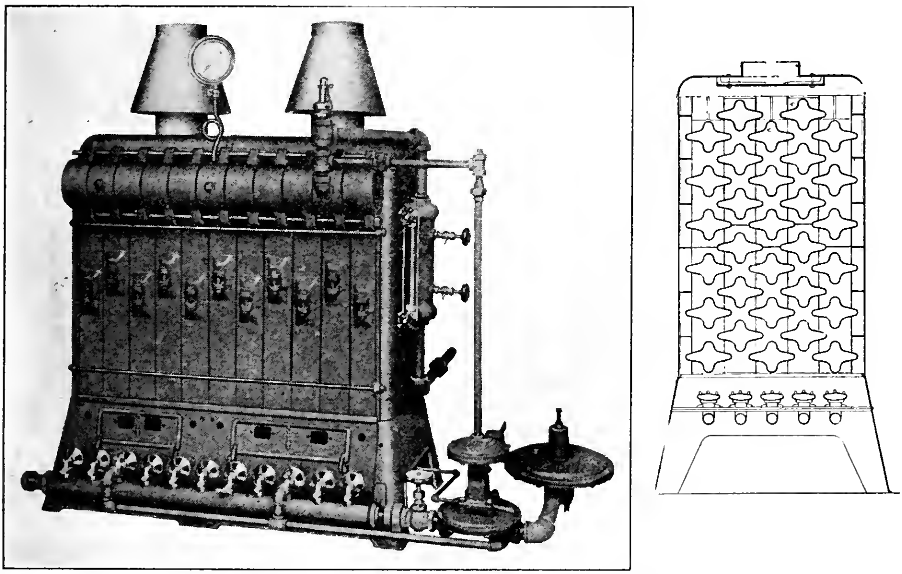

FIG. 4. A SPECIALLY DESIGNED GAS-FIRED BOILER, WITH SECTIONAL VIEW OF TUBE ARRANGEMENT AND FIRE TRAVEL

This shows that the percentage of CO_2 in the atmosphere of the room was 1.6 per cent with absolutely no ventilation. This condition would never exist in an ordinary dwelling for a period of four hours.

Another test was run to determine the amount of vitiation in the air under ordinary ventilating conditions. Readings were taken at four different points in the house, after the radiators had been burning for 4 hours, and were maintaining a constant temperature throughout the house. The following is the data collected:

Location	CO_2 per cent	O_2 per cent	CO	Room Temp. deg. fahr.	Outside Temp. deg. fahr.
1st floor	0.3	20.5	....	69	34
2nd floor, front........	0.3	20.0	....	70	34
2nd floor, rear.........	0.4	19.8	....	70	34
3rd floor	0.3	20.0	....	69	34

This test shows that the greatest amount of CO_2 existing in the house under ordinary conditions was 0.4 per cent.

There are several rules that we should never lose sight of when installing unvented heaters. First and above all, insist on an iron pipe connection; rubber tubing is a distinct life hazard and if used at all, it should be a specially approved tubing. Tubing may rot and deteriorate with age; it may be easily knocked loose from the heater by a child, a dog or by someone moving in the dark; or in a dozen other ways with possibly fatal consequences. The following extract quoted from the *American Gas Engineering Journal* of January 8, 1921, is very much to the point:

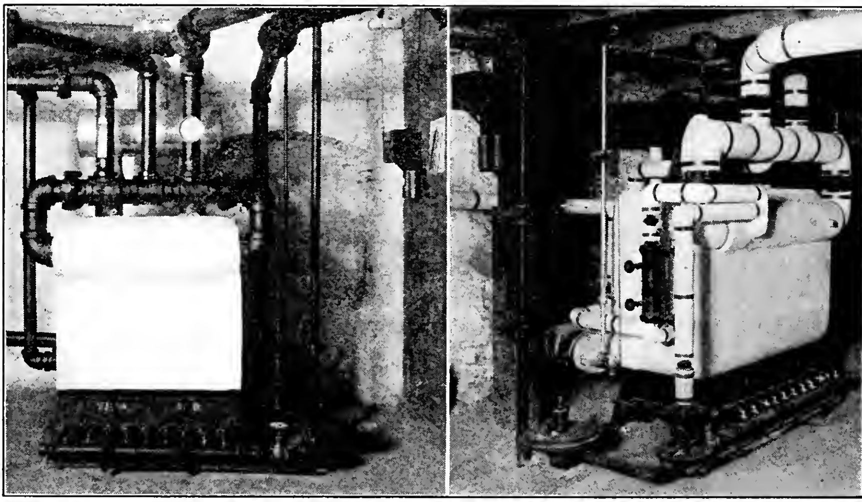

FIG. 5. TYPICAL INSTALLATIONS OF SPECIALLY DESIGNED GAS-FIRED BOILER, FOR HOT WATER AND STEAM HEATING

375 DEATHS BY GAS DUE TO LEAKY TUBING

Accidental gas asphyxiation was the cause of 375 deaths in New York City during the first eleven months of 1920, according to figures compiled by the office of Medical Examiner Norris. A conservative estimate of deaths in December, records for which had not been completed, would be twenty-five.

Most of the fatalities, according to George La Brun, compiler of the records, are due to defective rubber tubing attached to gas heaters. Eighty per cent of the deaths occurred in the winter months when heaters are in use.

"More than 50 per cent of these deaths could have been prevented by enactment of an ordinance which would compel the use of reliable tubing," Dr. Norris said.

The following is from the Chicago *Tribune* of January 5, 1920:

BIG DEATH TOLL LAID TO FAULTY GAS FIXTURES

Coroner bares results of investigation.

Faulty gas fixtures take an annual toll of several hundred lives in Chicago. During the winter an average of thirty deaths a month result from accidental asphyxiation.

"The majority of deaths in the accidental column were adults," the coroner said. "Usually, they had died because rubber connections between gas jets and heating or cooking appliances had rotted or sagged away from the ceiling fixtures allowing gas to escape."

We should also always remember that when the flame of a Bunsen burner flashes back into the mixing tube, combustion is incomplete and carbon monoxide is produced. Therefore, unvented room heaters with yellow-flame burners are preferable for most installations. We should never permit the installation of a Bunsen type unvented, in a bedroom.

The indiscriminate sale of cheap, improperly-designed and poorly-constructed gas heaters is a menace that should be curbed. They are sold by department stores and many dealers along with cheap tubing, with entire disregard of the conditions under which they are to be used, and to people totally ignorant of the first principles of how they should be connected. As long as this goes unchecked, they will occasionally provide work for the coroner.

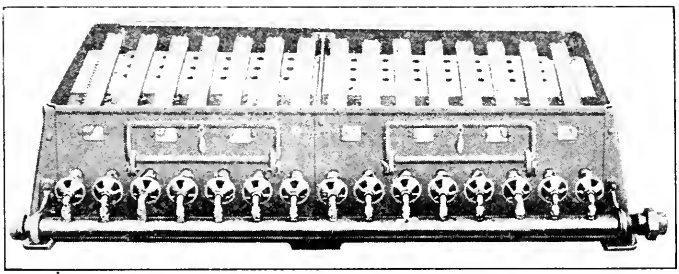

FIG. 6. ARRANGEMENT OF BURNER IN BASE OF SPECIALLY DESIGNED GAS-FIRED BOILER

In selecting unvented room heaters, we should, therefore, remember that they are all of the same efficiency, that is, 100 per cent, if they burn properly, although the gas log and the radiant and reflector types are more cheerful to look at and may help to warm one by the suggestive influence of a glowing fire. We should take care to see that the burner is well designed and not liable to flashback, if of the Bunsen type, and that the flame is not confined, or crowded in space, or brought into contact with any part of the heater, if of the yellow-flame type, also that we only place them where ventilation is ample, and above all, that we use iron pipe connections.

THE VENTED ROOM HEATER

In general, the vented heater is always superior to the unvented type if properly designed and installed. The most popular vented heaters are built in the form of standard steam radiators, either cast-iron or pressed-iron or steel, with the gas burning inside the sections. The products of combustion circulate through the sections, giving up their heat as they go, and are led to a vent pipe and so out of the building. Either natural or artificial draught may be used; each has its advantages and disadvantages. When natural draught is depended upon, we are somewhat limited and hampered in the placing of the radiators for we must select a place from which we can run our vent to a chimney or some other outlet. To secure success with natural draft, it is all important that the vent be adequate, direct, free of horizontal runs, unobstructed and free from back draughts. If this is provided, a well-designed radiator will operate with a loss up the flue of from 10 to 15 per cent. If the vent is not free and ample, the radiator will certainly give trouble; the burned gases will at times be forced back on the flame and into the room, and odor and bad atmosphere result.

When mechanical draught is used, a motor-driven fan is used to draw the burned gases from the radiators through a system of pipes, and

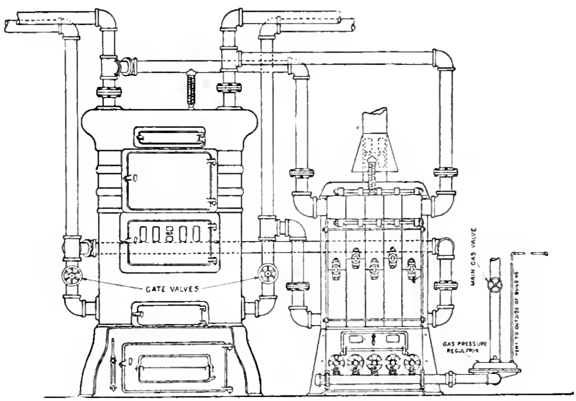

FIG. 7. TANDEM INSTALLATION OF COAL- AND GAS-FIRED BOILERS FOR HOT-WATER HEATING

expel them from the building. While this gives us more freedom in running our vent line, it introduces another complication by making a motor a part of the system, which will be disabled by motor trouble or failure of electric current.

The efficiency of the mechanically-vented system will be higher than with natural draft, but the electric current used and the greater installation cost must be considered in the comparison of the two. For many structures where it is possible to secure convenient vents, the vented radiator makes an admirable installation, especially where the need of heat is too scattered, or too intermittent or occasional to justify a central furnace installation.

CENTRAL GAS-FIRED HEATING PLANTS

This division of our subject represents its most interesting phase. The great majority of buildings of any size that are heated by coal are heated from a central furnace. Their owners are therefore particularly interested in a gas-fired boiler or furnace that may be substituted for their present coal-fired furnaces. The same owners are also usually interested in the question of converting their coal fired furnaces to the use of gas by the introduction of gas burners. We will deal with this phase of the subject first. I have had considerable experience in the conversion of existing coal-fired furnaces to the use of gas. I began by using ready-made burners such as are supplied by many firms in the natural-gas regions and were very extensively used in these regions when natural gas was very cheap. My attempts to apply these burners to artificial gas soon showed that a burner designed for natural gas cannot give entirely satisfactory results on artificial gas, especially if it is put under thermostatic control. It is almost sure to suffer from back-firing. In Baltimore, we very soon found that we could design better burners than we could buy. We also found that there was such an infinite variety of sizes and shapes of fire boxes among the furnaces we were called upon to convert that we gave up the attempt to use standard burners except for small, round fire-boxes and built our own burners, using wrought iron pipe for their construction, and designing each burner for its particular furnace.

The conversion of a boiler to the use of gas should only be attempted when gas is exceedingly cheap. Even with the best designed burners, properly placed, and when coal furnaces have a liberal amount of heating surface in comparison with the radiation supplied, and with the most careful baffling of the flues in the furnace and regulation of the damper in the flue line, the converted furnace will be much less efficient than a furnace designed and built to use gas only.

LOW FLUE TEMPERATURES POSSIBLE

A furnace can be obtained designed and built for gas, that will cool the products of combustion to a temperature within 20 deg. of the temperature of the water or steam in the furnace. This means a very small loss up the flue. With the converted coal furnace, however, we will have a flue-gas temperature from 100 to 200 deg. above the temperature of water or steam in the boiler, unless we are working with a very much over-size furnace. This means a high loss up the chimney. The only advantage of the converted coal furnace is that it is low in first cost and that the owner can easily return to the use of coal without large financial loss. However, our general experience will be that persons who can afford to heat with gas, can afford to pay for a gas-designed furnace.

Turning to the gas-designed boiler and furnace, we find a number of them on the market. Most of these have originally been designed for the use of natural gas. Some of these have redesigned their burners to make them operate properly on artificial gas. Some are attempting to handle artificial gas work without changing the burner design; these will probably give trouble on artificial gas.

The steam or hot-water boiler that will give and maintain satisfaction must have well-designed burners capable of lighting under practically all conditions with regard to pressure without back fire, as it is almost certain that a gas-fired boiler will be under thermostatic control. It must be efficient; the best boilers will show an efficiency of 80 per cent as a steam boiler and 87 per cent as a hot-water boiler under practical working conditions. It will be remembered that all the hydrogen in gas burns to water vapor and that this water must leave a steam boiler uncondensed, as the flue temperature must be higher than 212 deg. and that it carries with it its latent heat of evaporation. This will amount to about 7 per cent of the heat units in the gas, so that no steam boiler can possibly attain higher efficiency than 93 per cent, even though it cooled its flue gases to the temperature of the steam in the boiler and was so perfectly insulated that there was no radiation loss whatever.

With the hot-water boiler, a much higher efficiency may be attained; in fact, we could if we wished, approach very close to 100 per cent, but for several reasons, this is neither practical nor desirable. To attain a very high efficiency, we need only increase the heat-absorbing surface of the boiler per unit of gas burned and use more insulation, but beyond a certain point, the money so expended will cease to be wisely spent. The saving obtained in operating costs is not worth the cost of obtaining it. Also, if our efficiency is too high, our burned gases will be so low in temperature when they leave the boiler, that we cannot get them up the chimney and the flame smothers and floats. I would not want a boiler of more than 87 per cent efficiency, as in warm weather, it would give endless chimney trouble. I have seen cases where we have had to actually decrease the efficiency of a hot-water boiler to overcome such trouble.

PROPER VENTING OF GAS FURNACES ESSENTIAL

I cannot too strongly emphasize the importance of proper venting of gas furnaces, the more so because there seems to be a popular belief that they may be connected to any old vent, or even be operated with none at all. They may indeed be operated without a vent, but it involves a hazard we should never take. Every gas appliance operating under automatic control should be vented through a flue line of adequate size and decided pitch to a good chimney or some other outlet.

The consideration of the flue line brings up an interesting point. I have stated that the hot-water boiler may cool the products of combustion within 20 deg. of the temperature of water in the boiler. This means that the products in the flue line are often considerably below 212 deg. and this means a considerable amount of condensation of the water vapor produced by the combustion of hydrogen. There is a trace of sulphur in gas. This sulphur burns to sulphur-dioxide, which on being brought into contact with the water of condensation in the flue line, produces sulphuric acid. This acid will attack black or galvanized iron and I have seen a flue line destroyed in 3 months. My first attempt to overcome this trouble took the form of pickling black iron and dipping it in an acid resisting paint. Flue lines of this dipped iron lasted better, but sometimes went to pieces in one season. I then began to use terne plate which is sheet-iron covered with an alloy of 75 per cent lead and 25 per cent tin; this gave better results but sometimes failed to last two seasons. I then began the use of sheet lead for building lines and even in some cases where they were short, used terra cotta pipe. Both of these materials have disadvantages. The terra cotta is exceedingly heavy and clumsy to support, while the lead is soft and expensive. However, when a man puts in a line of either of these materials, he has something that will last indefinitely.

While time and space do not admit of a discussion of gas boiler design, a few words may be said of material. General practice has chosen the water-tube type and cast iron as the material. Many have thought that higher efficiency can be obtained by using copper tubes for water-heating work, but this view is not in accord with the real facts of the case.

While it is true that copper is a better conductor of heat than cast-iron, it is equally true that probably 85 per cent of the resistance to the flow of heat from flame and hot gases into the water, lies in the transfer of the heat from the flame to the metal. The resistance to the flow of the heat

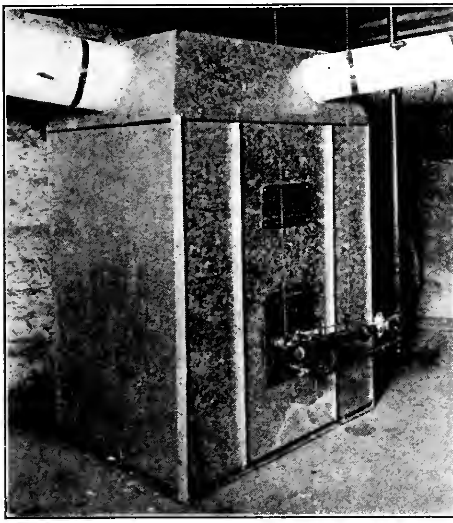

FIG. 8. SPECIALLY-DESIGNED GAS-FIRED FURNACE FOR WARM-AIR HEATING

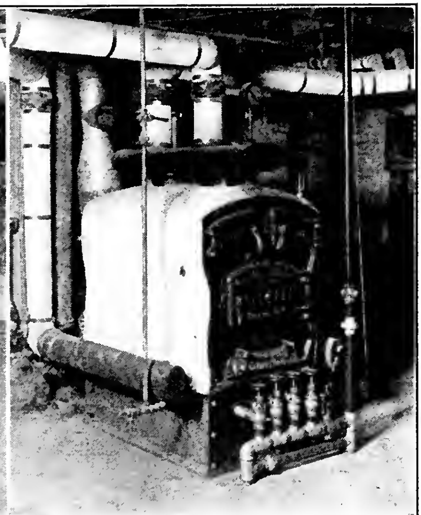

FIG. 9. COAL FIRED HOT-WATER HEATING BOILER CONVERTED FOR GAS FIRING

in the metal itself is very small in comparison. The cast-iron is as good, or better, an absorber of heat than copper; consequently by substituting copper for cast-iron, we decrease only a very small part of the total resistance to heat transfer. On the other hand, since cast-iron is cheaper than copper, we can give more heat-absorbing surface for a dollar than with copper, and when we consider ruggedness, resistance to corrosion and long life, it is seen that cast-iron is far superior to any material we can use to build our boilers.

The gas-fired central heating plant, when properly designed, and with burners that are proof against flash back trouble, used in connection with a reliable thermostat, is the ideal heating plant. It may be equipped with an automatic water feed in case of steam and vapor systems, and with an automatic low-water cut-off. It will then operate with practically no attention to the entire delight of the owner if he is willing to pay the cost of gas fuel.

COMPARATIVE COSTS

This brings us naturally to the relative cost of gas and coal. The true cost in any instance is, of course, not found by comparison of the bare cost of gas against the bare cost of coal. We must charge against coal, the value of the storage space it occupies, the cost of firing and attention to the furnace, the cost of handling, storing and getting rid of ashes, the damage done by dirt, dust and grit in the building, and the general worry and attention that are inseparable with coal-fired installations. On the other hand, gas must be credited with whatever value attaches to an absolutely automatic system, with the very valuable feature in business buildings, that it requires no attention over Sundays and holidays, that it may be shut off instantly and will not overheat the building in warm weather, and that it can be as quickly started again. I have known of cases where the rental value of the space made available by the use of gas practically paid the gas bill.

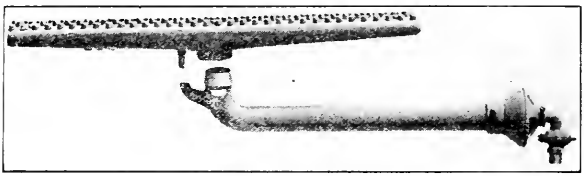

FIG. 10. A DESIGN OF BURNER AND MIXER FOR INSTALLATION IN A GAS BOILER

If we credit coal with 12,000 B.t.u. per lb., and compare it with gas containing 550 B.t.u. per cu. ft., we see that if the two fuels were burned with the same efficiency, it would require 48,870 cu. ft. of gas to do the work of a ton of coal. Our experience, however, shows that in a gas-designed boiler or furnace, compared against a good coal boiler or furnace, that 32,000 cu. ft. of gas is equivalent to one ton of coal when the whole heating season is considered.

The comparison of the cost against other fuels brings us naturally to the very interesting question of how to estimate the consumption of gas for heating a given building per heating season. The last heating committee of the *American Gas Association* has given us some interesting data on this subject, and we may approximately calculate the gas necessary to heat a building in any locality and under any conditions of exposure and gas supply from the following formula which applies to central gas-fired furnaces or boilers of fair efficiency:

$$T = \frac{A \times B \times C \times D}{E \times F \times G}$$

Where T = Total gas consumption for heating season;

A = Number of sq. ft. of standard direct cast-iron radiation in the installation under consideration;

$B =$ The heat emission value of one sq. ft. of standard direct cast-iron radiation which is 240 B.t.u. per hr. for steam, 200 B.t.u. for vapor and 150 B.t.u. for hot water;

$C =$ The average temperature difference for the heating season between the temperature maintained inside the building and the outside temperature;

$D =$ The number of heating hr. in the heating season, which are the number of hr. in the days during which the mean daily temperature is below 65 deg.

$E =$ An efficiency constant which we may assume at 0.75;

$F =$ Calorific value of the gas which is a local consideration;

$G =$ The design temperature difference which is the difference between 70 deg. and the local design temperature.

Knowing the gas rate in the given locality, we can quickly arrive at the comparative cost of the gas used per season against the cost of carrying out the same work with coal. Taking an average inside temperature of 65 deg., 180 days a season for heating, gas at 500 B.t.u., and design temperature difference at 70 deg. (values that are about right for a residence in the latitude of Baltimore), this formula gives 987 cu. ft. of gas per sq. ft. of steam radiation per heating season. Our experience with a large number of houses had caused us to adopt a figure of 1000. When gas is used in room heaters, the amount will be lower because of the more direct application of the heat.

The following percentages of total gas used by months for residences in a normal season, are interesting because they show how the gas consumption will be divided through the heating season: October, 5 per cent; November, 11 per cent; December, 19 per cent; January, 22 per cent; February, 20 per cent; March, 14 per cent; April, 7 per cent; May, 2 per cent.

THE FUTURE OF GAS HEATING

Gas is the ideal for heating houses. By its use we can make true every dream and realize every aspiration of modern man in his quest of the perfect heating plant. It is also a splendid thing from the view-point of the community. It does away with the hauling of coal and ashes through the streets, thereby reducing congestion and noise, while at the same time it tends to abate the smoke nuisance and reduce fire hazard.

From the standpoint of the conservation of our natural resources, it is ideal. When coal is burned in a furnace, we get nothing from it but heat, and in many cases, only a very small percentage of the heat units in the coal are efficiently used. If coal is gasified in a gas plant, we obtain gas from heating, coke for metallurgical processes or further gas making, ammonia for fertilizers, benzole and toluol for motor fuel, and tars for road making, dye and medicinal purposes.

The only source of heat that can claim to excel gas from the utilization point of view is electricity, and since the best and most efficient steam-driven electric plants are only able to put about 16 per cent of the heat units in the coal into electric energy, while the gas plant can put from 75 to 80 per cent, electricity as a source of heat must cost three or four times as much as gas when they are both derived from coal.

From the above consideration, we see that the only drawback to the adoption of gas heating on a most extensive scale is that of cost. The figure of 32,000 cu. ft. to a ton of coal, with gas at $1.20 per 1000 cu. ft., gives an equivalent cost of $38.40, and this is a fair comparison for the central heating plant. If coal costs $14.00 per ton, the consumer must see about $24.00 worth of savings and advantages in gas to justify its use.

In special cases, and for auxiliary use, the comparison is much more favorable to gas.

The heating load is not entirely desirable business from the point of view of the gas company. It has in fact, very serious disadvantages. It is only half-year business and has very little diversity factor. The weather affects all alike; consequently, the company must have enough capacity in its generating plant and distribution system to meet the coldest day and the coldest week in the year. A large proportion of this investment will be idle and unproductive during 11 months of the year. This condition is aggravated by the fact that in many cities the winter already exceeds the summer output.

NEW STANDARDS HELPFUL

The gas companies are just being freed from the obsolete and oppressive candle-power standards, but these standards have retarded their development toward the supplying of large heating loads. The use of oil must be eliminated from gas manufacture before a cheap gas can be produced. Rates must be stable and adequate before capital can be obtained to make plant and distribution system additions to supply large blocks of heating business.

These considerations seem to fairly indicate that gas is destined to be the fuel of the future, but its advent will be very gradual. Before this can be attained, methods of manufacture must be adopted which will turn all the coal used into gas and by-products in the most economical manner, and the use of oil must be eliminated. This will mean the production of a gas of considerable lower heating value per cu. ft. than at present. I believe that the gas of the future will have a heating value of somewhere in the neighborhood of 400 B.t.u. per cu. ft., and this will be adopted because it will be the gas in which we can give the customer the greatest number of heat units for a dollar. This gas will have a higher flame temperature and can be more efficiently applied, than either natural gas or the rich candle-power gases such as were formerly furnished in the artificial field with heating values of from 600 to 650 B.t.u. per cu. ft.

At present, there is a great demand and a great field for gas heating, but it must be applied with judgment, and after careful consideration of the conditions affecting each case. For some buildings, it is very difficult to get coal into the buildings or to provide a storage space for it. For other buildings, the heating is intermittent, or only certain portions of the buildings, possibly the upper floors, are to be heated. In many cases, when all things are considered, it may be profitable to use gas. In others, especially residences, the owners of considerable means will be glad to pay the additional cost of gas to obtain the luxury of a perfectly heated house and automatic control.

The whole trend and development of modern life encourages the use of gas. The difficulty of obtaining and retaining domestic help and janitors, the desire to eliminate coal storage space and to apply basements to other uses, the instinct to get rid of dirt and dust, the tendency to use automatic devices, the economy of time and effort, the multiplication of comfort and the increase of efficiency and fuel conservation. These are all served by gas heating and the people will demand it. It is the duty and problem of heating engineers to so direct its application that the interest of both user and gas company may be best protected and advanced for they are mutual and must move together.

TABLE 1. COST OF 100,000 B.T.U. AS CONTAINED IN VARIOUS FUELS

Fuel	Heat Content	Price	Cost per 100,000 B.t.u.	
Soft Coal	13,000 B.t.u. per lb.	$8.00 per ton	$0.027	
Hard Coal	11,500 " " "	15.50 " "	0.060	
Coke	12,000 " " "	16.00 " "	0.099	
Charcoal	13,000 " " "	0.50 " bushel	0.255	
Wood	6,500 " " "	20.00 " cord	0.154	
Fuel Oil	19,500 " " "	0.09 " gal.	0.068	
Gasoline	20,000 " " "	0.32 " "	0.214	
Natural Gas	1,000 " " cu. ft.	0.40 " M. cu. ft.	0.04	
Producer Gas	140 " " " "		0.10	Large producer unscrubbed gas
Steam	1,000 " " lb.	0.90 " M. lb.	0.09	
Electricity	3,412 " " K.W.H.	0.04 " K.W.H.	1.17	
City Gas	500 " " cu. ft.	1.00 " M. cu. ft.	0.20	

The above table shows the relative cost of the most widely used fuels when considered on a basis of the heat units they contain and at the price per unit of fuel. These unit prices, however, are at present subject to considerable fluctuation so that in using the table it should be corrected, if necessary, for current prices. The table takes no consideration of the efficiency of application or the cost of handling.

TABLE 2. COMBUSTION DATA ON GASES

	Ml. wt.	Sp. gr.	B. t.u. per cu. ft.	Lb. Air to burn lb. gas	Cu. ft. Air to burn lb. gas	Vol. Air and gas mixture	B.t.u. per cu. ft. mixture
H_2	2	0.0692	348	34.56	2.38	3.38	103
CO	28	0.967	349	2.46	2.38	3.38	103
CH_4	20	0.553	1065	17.28	9.54	10.54	101
C_2H_2	28	0.898	1551	13.3	11.9	12.9	120
Natural Gas		0.55	978		9.28	10.28	95
Pittsburgh Coal Gas		0.42	646		5.78	6.78	95
Car Water Gas		0.54	575		5.02	6.02	95
Blue Water Gas		0.53	295		2.25	3.25	91
Producer Gas		0.81	144		1.05	2.05	70
Blast Furnace Gas		1.02	.91		0.67	1.67	54

No. 605

VENTILATION STUDIES IN METAL MINES

By D. Harrington,[1] Denver, Colo.

Non-Member

WHILE ventilation has practically always been deemed an integral part of coal mining, metal mines have rarely paid much, if any, attention to air circulation until forced to do so by occurrence of some untoward condition or accident. Yet metal mines actually have as great need of efficient circulation of air as have coal mines. The coal mine must remove the dangerous explosive gas methane, also fumes from explosives and in occasional places other gases such as CO_2 or nitrogen; metal mines have greater necessity to remove fumes from explosives, frequently have occasion to remove CO_2, nitrogen and other gases from strata and even the coal miners' explosive gas *methane* is occasionally encountered; in addition, circulating air currents are urgently needed in metal mines to reduce excessively high humidity and temperature so frequently found in our metal mines but very rarely in coal mines. The immense quantities of minutely fine particles of rock dust floating in the stagnant air of metal mines which are very largely responsible for miners' consumption and other diseases so prevalent among our metal miners in many regions, could be almost wholly removed by adequate ventilation. The generally accepted conclusion that coal miners have a healthful occupation and live to a ripe old age, and that many metal miners contract diseases such as lead poisoning, miners' consumption, etc., and either die early in life or are incapacitated in middle age, is due almost wholly to the superior working conditions in coal mines, chiefly ventilation.

Our larger metal mines are readily comparable in many respects to an immense office building or a hotel. The various levels correspond to the floors of the buildings, except that in a mine they are 100 or more feet apart vertically and in a building 10 to 20 feet; the drifts and crosscuts correspond to the halls and corridors; and the raises, stopes and other working faces to the offices, sleeping rooms, etc., with the difference, however, that in the mine there are no openings to correspond to windows to the outside, and only too frequently no openings between working places corresponding to interior doors between offices or rooms, thus leaving only one opening to the working place. In general, the present practice even in the better ventilated metal mines is to cause currents of air to flow along the main drifts and crosscuts corresponding

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[1] Mining Engineer, United States Bureau of Mines.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, O., June, 1921.

to halls and corridors of a building, with only such amounts of air going into the working places as might be expected to seep by diffusion into the rooms of buildings if all outside doors and windows were tightly closed and the one door to the hall left open. The workers are usually engaged in some occupation, such as drilling, timbering, shoveling, etc., which fills the air with fine dust (the most dangerous kind), or possibly they set off some dynamite to bring down ore, liberating not only clouds of fine dust but also of smoke, laden with poisonous gases; and only too frequently the surrounding walls are damp and have temperature 80 to 100 deg. fahr., or even more, and the stagnant air of the place has temperature of 80 to 100 deg. fahr. with a relative humidity of 90 to 100 per cent.

The metal mine underground worker frequently has a place in which he works daily under conditions analogous to those on the hottest, stillest, most humid day which can be found in any of our large cities and, in addition, he frequently has to breathe air laden with fine dangerous dust and with various quantities of poisonous gas, such as CO or oxides of nitrogen from explosives. The underground worker in some metal mines must pause several times daily to wring perspiration out of the few pieces of clothing he wears and even pour perspiration out of his shoes from time to time. The ordinary condition of the underground metal mine working face often is similar to that which would be produced should all the windows of a sleeping room be closed tightly and air derived wholly from open door to the hall, the resultant headache to sleepers from such situation being augmented to the miner by dust, gases from explosives and decaying timber, and possibly by high temperature and humidity and alleviated to a slight extent by release of compressed air.

MINE INVESTIGATIONS

Working under general direction of G. S. Rice, chief mining engineer of the U. S. Bureau of Mines, data on metal mine ventilation has been obtained by personal observation of the writer on this specific subject in over fifty mines in ten states during the past five years, supplemented by general underground observations during the past 20 years in over 150 coal mines in 15 states, and in over 100 metal mines in 14 states. In addition, the writer has had access to reports by various other investigators both within and outside of the U. S. Bureau of Mines along this or kindred subjects.

In a detailed study of a large metal mine in which much better than average ventilation practice was in vogue and workings extended almost from the surface to below the 3000-ft. level, data were obtained at 158 working places, of which 87½ per cent were below the 2000-ft. level, 35 per cent below the 2500-ft. level, and about 75 per cent of the observations were at actual working faces. It was found that average wet-bulb temperature of the 158 places was 76.8 deg. fahr. and average relative humidity 91.3 per cent. There was sufficient air movement to be detected by ordinary observation in 42 out of the 158 places, and an additional 29 places had enough movement to be detected through very slight inclination of a candle flame leaving 87 out of 158 places (or over 55 per cent) with not the slightest detectable movement of air.

The wet bulb temperatures were as follows:

40	places	under 70 deg. fahr.
45	"	70 deg. to 80 deg. fahr.
61	"	80 deg. to 85 deg. fahr.
12	"	over 85 deg. fahr.

The relative humidities were as follows:

22	places	under 85 per cent.
12	"	85 to 90 per cent.
51	"	90 to 95 per cent.
73	"	over 95 per cent.

In general, the places with the high wet bulb and high relative humidity and little or no air movement were those where men were doing the hardest and most important work, that is, at the working face, drilling, mucking, or shovelling, timbering, etc.

A comfort rating was given each place by the writer while in the place, the result being compiled as follows:

78 places—with average wet bulb 72.1 deg. fahr. and relative humidity 88.2 per cent—were rated comfortable and 80 places—with average wet bulb 81.4 deg. fahr. and 94.2 per cent relative humidity—were said to be uncomfortable. Out of 85 places with wet bulb less than 80 deg. fahr., 73 were rated comfortable irrespective of air movement or relative humidity; uncomfortable rating was given places with wet bulb less than 75 deg. only if there should be excessive dust or gas present or dry bulb temperature should be very high—say up to 100 deg. fahr.—and in the latter case if the air was moving 500 ft. per min., or over, the place was rated comfortable. In this mine about 400 to 450 men were employed underground per shift and total quantity of air supplied was about 60,000 cu. ft. per min. Underground rock and water temperatures varied from about 60 deg. fahr. near the surface to 104 deg. in the lowest levels.

In Table 1, there are compiled interesting data as to ventilation features in nine fairly large mines located in two western states, one of which has hot and dry climate, the other cold and somewhat damp climate. It is noticeable that average wet bulb of all underground places visited in every case lies between 70 deg. and 80 deg. fahr., and the relative humidity is 85 per cent or over.

Mines 1 and 2, with average wet bulb of all places visited 72.5 deg. and 72.3 deg. fahr. respectively, had but 14 and 25 per cent respectively places rated uncomfortable, though 96 per cent of places in Mine 1 and 100 per cent of places in Mine 2 has relative humidity over 85 per cent. It is equally noticeable that all mines with wet bulb 75 deg. fahr., or over, had over half of places rated uncomfortable. These two facts seemed to indicate that 75 deg. wet bulb was the upper-most wet bulb average at which comfort can be expected irrespective of humidity, movement and possibly dry bulb temperature. It is also noticeable from this tabulation that relative humidity of 85 per cent, or over, was found in more than 60 per cent of all places visited in all the mines in the table, and in 75 per cent, or over, of the places of all mines but one. Air movement was practically or wholly lacking in over 60 per cent of all

TABLE 1. CONDITIONS OF AIR IN VARIOUS METAL MINES

Mine	No. of places	Wet bulb deg. fahr	Rel. Hum. per cent	Comfortable places per cent	Uncomfortable places per cent	Wet bulb over 75 deg. fahr. per cent of places	Rel. hum. 85 per cent or over per cent of places	Air Movement Little or none % of places	Season	Climate		Average Mine Depth
										Temp.	Hum.	
1	28	72.5	94.7	86	14	14	96	32	Summer	Hot	Dry	About 600 ft.
2	20	72.3	93.2	75	25	5	100	40	Summer	Hot	Dry	" 600 "
3	61	75.5	89.0	42	58	54	89	46	Fall	Medium	Medium	" 1800 "
4	28	78.3	90.0	25	75	82	75	25	Winter	Mild	Humid	" 1600 "
5	16	78.0	93.0	31	69	87	100	44	Winter	Mild	Humid	" 1200 "
6	11	75.1	94.0	37	63	64	91	91	Winter	Mild	Humid	" 1200 "
7	158	76.8	91.3	49	51	74	86	73	Winter	Cold	Dry	" 2500 "
8	87	77.8	85.2	32	68	76	68	75	Summer	Medium	Medium	" 2200 "
9	77	74.5	87.7	*	*	60	75	64	Summer	Medium	Medium	" 2200 "

NOTE: *Not rated as to comfort.

places visited in four of the nine mines and in over 25 per cent of all places visited in all nine mines, these places with stagnant air being almost invariably the working faces. The nine mines in the tabulation employed a total of about 5000 men underground during three shifts, and of the nine mines tabulated five have extensive workings, are heavy ore producers and are known as being of the decidedly better ventilated class.

CONDITIONS AFFECTING METAL MINE VENTILATION

It appears that in order of their importance the main features affecting air in metal mines are: 1. Movement. 2. Temperature. 3. Relative Humidity. 4. Gases. 5. Dusts.

The matter of movement, or velocity, of air is by all means the most important consideration in effecting adequate ventilation of mines and in the protection of the health of and the forwarding and maintaining of the working efficiency of the underground workers.

Temperature of underground air is affected by outside air temperature in varying degree, dependent on depth and extent of workings, air velocities and other considerations; temperature of mine air is very definitely affected by underground rock and water temperatures, by quantity of air flowing, by oxidation (or decay) of timbers and ores, and by mine fires. It is also affected to a greater or less extent by friction due to velocity of flow, by moving of ground, firing of shots, heat from lights used and from breathing of animals. Heated air obtained from other mines and from electric motors and other machinery may seriously affect the local temperature of air of underground workings.

Relative humidity of underground air is affected to some extent by humidity of surface air, but much more vitally by moisture content of walls of underground workings and especially by water dripping through the air. Quantity, temperature and velocity of air flowing also ultimately effect the humidity of underground air. Handling of air by small fan and pipe units also locally affects mine air humidity and may be utilized to govern humidity of underground air to a certain extent.

Gases to be found in mine air may come from surface air, from breathing of men and other animals, from lights used, from firing of explosives, from compressed air used with machines or as blowers, from operation of various kinds of machinery and from various rock or other strata encountered, as well as from mine fires that are active or incipient.

Dusts found in metal mine air are largely derived from dry drilling, from blasting, from shovelling or mucking, from tramming or dumping ore, timbering, etc. Probably well over 50 per cent of all metal mining has siliceous material in ore or containing walls, hence has siliceous dust which as far as present-day knowledge goes, is the most dangerous of all dusts, especially when taken into the lungs in extremely finely divided form such as dry drilling, blasting and mucking throw into the air. Certain siliceous dusts seem to have far less injurious effects than others of essentially the same composition. Dusts of other than siliceous character, while thought not to be so definitely and immediately dangerous, are nevertheless likely to be harmful ultimately; as for example, the dusts of certain soluble lead ores which affect workers by skin absorption as well as through breathing.

GENERAL CONCLUSIONS

Below are some general conclusions of the writer at the present time given with full knowledge that future investigations may force modification of some of these views:

1. In mining, it appears that ventilation, fire protection and prevention, health, safety and efficiency are very closely interlocked.

2. There is equal reason for providing adequate ventilation for most metal mines, as for providing ventilation for coal mines.

3. Metal mines rarely, if ever, make provision for ventilation until forced to do so by some untoward condition or occurrence; coal mines on the other hand universally provide for ventilation.

4. Efficient ventilation of metal mines consists in supplying at all times such volume of circulating air at places where men work as will enable the worker to exert himself in comfort at maximum physical capacity without endangering his health.

5. Many metal mining operating officials are ignorant of the principles of air circulation, this being true of those technically educated as well as of those without technical training.

6. Workers in metal mines, including shift and other bosses, must be educated to respect ventilating devices such as doors, regulators, overcasts, brattices, fans, etc., as coal miners do, and to become as familiar with those devices as coal miners are. Many present-day *hard-boiled* metal miners and bosses consider ventilation a useless fad and obstruct rather than aid ventilation improvements.

7. Ventilation should be under definite constant supervision, and preferably the person in charge should report to the highest officials as many local officials in metal mines are not in sympathy with ventilation betterments.

8. Each mine should be ventilated wholly within itself; interventilation of mines is likely to be dangerous, inefficient and unsatisfactory.

9. Every mine, coal or metal, should have mechanically driven fans placed on the surface, fireproof housed and capable of reversing air currents with minimum delay. Metal mines should provide fan ventilation from start of opening in order to avoid dangers from explosives fumes, etc., and provide fresh air to workers.

10. At seasons of the year when the temperature of surface air and of underground rock and water are about equal, mines relying on natural ventilation have periods when air circulation is sluggish, or ceases utterly, or reverses in direction.

11. At time of mine fire, naturally ventilated mines are likely to be at a decided disadvantage through inability to control direction of air currents.

12. There are records of naturally or otherwise inadequately ventilated mines filling with CO_2 or other gas, overcoming some of the workers and compelling suspension of work for considerable periods of

time. Upon establishment of efficient mechanical ventilation this situation was readily controlled.

13. Workers in many metal mines are much less healthful than workers in coal mines, due largely to the superior ventilation of the collieries.

14. Miners' consumption, the scourge of metal miners, is caused primarily by breathing of very fine particles of certain mineral dusts and especially of siliceous dust. Over 50 per cent of our metal producing mines are working in more or less siliceous material, and through lack of ventilation, giving this dangerous dust maximum opportunity to exercise its harmfulness. Lead poisoning afflicts workers by skin absorption of soluble dust of lead ores as well as by breathing of such dusts.

15. The best remedy for the dust menace in mines other than preventing its formation, is the universal coursing of currents of air to remove the dust, as it has been proven that the very fine most dangerous dust in metal mines remains suspended in still air several hours.

16. Metal mine dust acting through miners' consumption, lead poisoning, bronchitis, etc., is the chief instrumentality in perhaps more deaths annually among our 175,000 metal miners than coal dust causes to our 700,000 coal miners through explosions.

17. It is fairly well established that miners' consumption is caused chiefly by siliceous dust, but it is probable that any kind of dust in mine air will prove harmful to workers ultimately. Investigations in our metal mines indicate that the air of mines so far studied is from seven to over 40 times as dusty as in South African gold mines.

18. Intake air of metal mines is frequently dusty through having crusher house or other dust producer near collar of intake air shaft, or by having ore skips or cars or ore dumping places in intake air passages without slightest attempt to allay dust or prevent its entering the mine.

19. The dustiest, most unhealthful occupation underground is dry drilling, and the average dust content of air of places doing this work in five large mines in various parts of the United States was 205 milligrams per cubic meter of air, yet for similar work in South African mines, but using available precautions against dust formation, the dust content of air is less than 5 milligrams per cubic meter of air.

20. While there are regulations in several states to prevent dust formation in drilling, these regulations are not always lived up to. Miners, while recognizing the dangers from dust, often prefer to take the risk rather than endure the slight discomfort or extra trouble in using the precautionary methods or devices; and mine and state officials appear to feel that unless the miner will willingly aid in protecting himself, they can not force him to protect his own health and incidentally that of his family. It is the writer's opinion that dust prevention devices of proven success, such as present-day self-rotating wet stopers should entirely supplant dry drills, and their use be enforced upon both miners and operators in metal mines, as there is absolutely no valid excuse for dry drilling in present-day metal mining except in a very few instances.

21. Spraying devices available to reduce dust while drilling may be effective if used intelligently; on the other hand, they may even intensify the air dustiness if used without intelligence and unfortunately the latter

is generally the case. Efficient water drills are now available for all purposes in metal mining (including efficient wet stopers for upper holes) and dry drilling should be prohibited.

22. Some metal mines with high dust production under certain conditions at working faces, have comparatively low dust content of the air of these places at other times, and low average dust content of all places due to efficiency of ventilating currents especially at working places. Significantly these mines appear to be singularly free of miners' consumption or other diseases, yet the employees work at top-speed and the material handled is highly siliceous.

23. The use of poorly placed compressed air hose blowers at working faces frequently intensifies air dustiness by allowing high velocity compressed air streams to pass through dry, loose, finely divided ore or other material being drilled, shovelled, etc., and forcing workers to breathe this air highly impregnated with dangerous fine dust.

24. While finely divided dust in mines is probably the *chief* cause of miners' consumption, it is now recognized that there may be other factors of almost equal influence, such as high temperatures and humidities, harmful gases and lack of air movement; all of these readily remedied by ventilation.

25. It appears that with dry bulb temperature below 75 deg. fahr., mine working places may be comparatively comfortable irrespective of air movement or relative humidity. However, the presence of air heavily depleted of oxygen or impregnated with gases such as CO_2, CO, oxides of nitrogen, etc., may produce uncomfortable or unsafe conditions; also such places may be both uncomfortable and unhealthful if large quantities of finely divided dust are present.

26. With dry bulb air temperatures above 75 deg. fahr., comfort and maximum working efficiency can be attained only when the air is moving, this being especially true if the air has high relative humidity. The exact velocity necessary is a variable dependent upon the temperature and humidity. This subject is to be further investigated.

27. Saturated atmospheres, up to nearly blood temperature, may be made endurable and even to a considerable extent comfortable by providing sufficient velocity. Some experimental work has recently been done on this by Dr. Sayers, Chief Surgeon of the U. S. Bureau of Mines, and the writer, and detailed report is soon to be issued.

28.[1] In still air in metal mines with about 85 deg. fahr. and 90 to 100 per cent relative humidity, there was little effect on persons completely at rest, but upon doing even moderate work body temperature rose to over 100 deg. fahr., blood pressure fell perceptibly and pulse beat rose materially. In still air with temperatures 90 to 100 deg. fahr., and about 90 per cent relative humidity even when practically at rest, body temperature rose quickly reaching over 102 deg., blood pressure fell rapidly, pulse beat increased abnormally and was very sensitive to even slight exercise, perspiration was very profuse and dizziness, physical weak-

[1] A Preliminary Study of the Physiological Effect of High Temperatures and High Humidities in Metal Mines, by R. R. Sayers and D. Harrington, U. S. Public Health Reports, Jan. 28, 1921.

ness, mental sluggishness, and headache were found; upon attempting even light work these symptoms were greatly augmented.

29. Relative humidity, even up to the saturation point, does not appear to be harmful to health, comfort or efficiency until temperatures run above 75 deg. fahr.; and if sufficient movement is supplied, high relative humidity is not particularly harmful until temperatures are well over 90 deg. fahr.

30. With exception of blind end working faces, metal mine air, in general, is not particularly deficient in quality. However, blind end faces of drifts, crosscuts, raises, winzes and stopes in metal mines are likely to have air deficient of oxygen, and high in nitrogen or CO_2, and possibly in CO, oxides of nitrogen or other impurities. There are many records of asphyxiation in metal mines from these gases.

31. Practically all of the explosives used in metal mines give off small percentages of poisonous fumes—CO, H_2S, or oxides of nitrogen—when fired, and when these fumes are not removed from confined working places by ventilation they cause headache, nausea and may even cause death. The gelatin dynamites give off less quantity of dangerous gases than do the ammonium dynamites and the latter give off much less than the straight nitroglycerin dynamites, hence the straight nitroglycerin dynamites should not be used underground; and all places using any kind of explosive should be thoroughly ventilated after blasting and before workers arrive. Good ventilating currents should also be provided while shovelling blasted ore at the face to prevent workers from being affected by headache, nausea or other illness from breathing explosive fumes stirred out of muck piles. Compressed air hose blowers are not particularly effective to remove explosives fumes.

32. In stagnant air comparatively small quantities of impurities, such as 0.30 per cent, or over of CO_2, 0.02 per cent or over of CO or oxygen below 20 per cent, give headache, dullness, etc., and this is particularly true when temperatures are above 80 deg. fahr. However, these small quantities of impurities are not noticeable when there is perceptible movement to the air.

33. Frequently blind end working faces in metal mines have air so depleted of oxygen that a candle will not burn and carbide lamps must be used, hence oxygen is below 18 per cent and CO_2 runs several per cent; occasionally entire mines are found with this condition which many metal mine managers held to be perfectly all right. There is absolutely no question that men working in atmosphere which will not support combustion of a candle can not deliver maximum efficiency and their health must suffer ultimately.

34. Mines with cool working places which allow men to work at top speed, especially when contracting, are likely to be extremely dangerous to health of workers unless provision is made to remove explosive fumes and fine dusts from working places by ventilating currents.

35. Mines with high temperatures, above 75 deg. fahr., and high humidity, above 85 per cent, are likely to lose from 25 per cent to as much as 75 per cent of the efficiency of workers, and workers are likely to become unhealthful ultimately unless moving currents of air are sup-

plied working places. Unhealthful and inefficient results are hastened and intensified if fine dust is present, especially siliceous dust, and if there is much blasting done especially when men are in the mine. A mining man in charge of a mine with hot workings over 80 deg. fahr. in siliceous ore and with practically no ventilation, states that he has noticed that a number of men have died within a few years after having worked any considerable length of time in a certain hot, dusty section of his mine, there being practically no air circulation.

36. Many accidents in metal mines are due to deficient ventilation; failure to remove smoke and fumes from explosives prevents proper inspection of working places to make them safe and, in addition, many men have been asphyxiated in explosives fumes; moreover, in hot, humid, stagnant air men are likely to be affected by dizziness or by lack of ability to think clearly or quickly, or they may faint at an inopportune time and be killed. There are instances where men have been known to drop dead due to heart failure in these hot places.

37. In general, the giving of air movement or velocity at working places is of more importance than any other consideration in providing adequate metal mine ventilation.

38. While variations in surface air temperature and humidity may have noticeable effect on mines with shallow workings, in general, they change conditions very little if at all at faces in mines with extensive workings, hence underground working temperatures in large mines vary little, if at all, due to outside air circulation.

39. Air flowing in underground passages rapidly takes the temperature of the surrounding rock; the rate of change is variable and frequently is as high as, or higher than, 1 deg. fahr. for every 100 ft. of travel even when air velocity is several hundred ft. per min. Still air underground rarely varies more than a few degrees from temperature of surrounding rock or water.

40. Rock temperature generally increases with depth, the rate of increase varying from 1 deg. fahr. or more, per 100 ft. of depth in certain districts of the western part of the United States, to but ½ or 1/3 deg. fahr. per 100 ft. of depth in other regions, both of the United States and of foreign countries. In Montana, rock temperature is copper sulphide veins is about 108 deg. fahr., at a point 3800 ft. below the surface, rate of increase being about one deg. per 100 ft. of depth. In a lead sulphide vein in the Coeur d'Alenes in Idaho, rock temperature 2000 ft. below the surface was but 50 deg. fahr. In a Michigan copper mine with native copper ore, rock temperature was about 82 deg. fahr. at a point 5000 ft. below the surface. In a gold-bearing quartz vein in Arizona, rock temperature was 90 deg. fahr. at a point 600 ft. below the surface, while in a quartz deposit bearing copper sulphide in another Arizona district, rock temperature 600 ft. below the surface was but 70 deg. Temperature of coal in place in coal mines in the United States is rarely above 70 deg. fahr., and is generally much lower. Recent magazine articles give the rock temperature of the Kolar Gold Field in India at 118 deg. fahr. at a point 6100 ft. below surface, and 98 deg. fahr. at the 4000-ft. level of the St. John Del Rey Mine in Brazil; while

it is calculated that at the 8000-ft. level of City Deep Mine in South Africa the rock temperature will be but about 97 deg. fahr.

41. Rock temperature may vary at the same depth in different kinds of material; a copper sulphide ore with quartz gangue had rock temperature several degrees higher than a zinc sulphide ore in quartz gangue in a parallel vein about 200 ft. distant both on the same level, and both practically free of water.

42. Water standing still or flowing on floor in mines readily communicates its temperature to surrounding air; water dripping through the air quickly brings the air practically to temperature of the water drippers, and profuse water drippers will determine temperature of air irrespective of rock temperature. Water temperature underground is generally the same as that of surrounding rock but this is not always the case.

43. Mines having rock temperatures, thence generally air temperatures, above 80 deg. fahr., frequently have available water piped from the surface and found underground with temperature below 70 deg. fahr. Few, if any, mines take advantage of this cool water to cool the air by installation of sprays, yet this is a very effective method of cooling air. Mine managers state that they fear that water sprays will cause excessive humidity, forgetting that mines generally have the humidity anyway, and that if the air can be cooled to 75 deg., or below, and given a slight movement the high humidity is not harmful. Recent physiological experimental work in South African mines[1] shows that men working in stagnant air with relative humidity 95 per cent and temperature 87 deg. fahr. increased the amount of work performed 46 per cent by the mere expedient of installing a small fan to move or stir the air, showing that high humidity in itself is not particularly detrimental, at least until temperatures are well above 90 deg. fahr.

44. Air passing through fans frequently has dry bulb temperature increased several deg. fahr. and relative humidity automatically decreased. In one instance, an underground fan with air delivery over 20,000 cu. ft. per min. had 8 deg. higher temperature of air at delivery than at intake, the points of measurement being less than 50 ft. apart. Similarly small fan-canvas pipe units used underground for local ventilation frequently have delivered air several degrees higher temperature than that of intake air at the fan.

45. Small electrically driven fans with galvanized iron or canvas tubing are being rapidly introduced into metal mines to carry air to dead-ends. The galvanized iron has the advantage of allowing of reversing of air currents to pull smoke out after blasting, then to force moving air to workers after removal of the smoke; moreover, it does not decay as fast as the canvas. The canvas tubing must be, in general, used only in forcing air to the face, its advantages being low first cost, readiness of installation and removal, flexibility in conforming to bends or turns, and ease of repair; moreover, because of its readiness of installation and removal, the canvas tubing can be brought close to the working face at ordinary times and easily removed prior to blasting to prevent its destruction. Either method readily admits of placing from

[1] Paper by A. J. Orenstein and H. J. Ireland, in *The Journal of The South African Institution of Engineers,* March, 1921.

500 to 5000 cu. ft. of moving air per min. at the working face at comparatively small cost.

46. Compressed air from end of air hose is used to a very great extent to ventilate hot, stagnant blind end working faces in metal mines. These blowers deliver about 100 cu. ft. of air per min. and its temperature rarely varies much over 2 or 3 deg. from the temperature of the rock and air of the working place. Such compressed air blowers are inefficient as to removal of smoke or gases, provide comparatively little pure air and give very little reduction in the temperature of the surrounding air. Moreover, it costs about 100 times as much to place 1000 cu. ft. of compressed air at a working place as to circulate a like amount of air by ordinary ventilation methods.

47. The use of electrical machinery underground locally causes considerable increase of air temperature. A recent magazine article describes a proposed fan installation at a South African mine to force 75,000 cu. ft. of air per min. from the surface into a mine for the sole purpose of ventilating the region around an electrically driven hoist, this air to be removed from the mine after passing through the hoist room.

48. Cooling of air in mines is effected by use of sprays of cool water, by refrigeration, by rapid coursing of air brought from the surface, or carried through workings with cool walls, and by excluding air from abandoned workings and return air from currents of active workings. The method of water sprays is not employed nearly as much as it should be; the method of refrigeration is costly and found only in a mine in Brazil, as far as the writer's information goes; and the rapid coursing of air currents from the surface which can be brought about most efficiently by establishment of definite separate splitting systems is used only occasionally though it is quick, cheap and efficient. Failure to seal abandoned places having decaying timber and hot rock or water sends much unnecessary heat into metal mine air, and re-using of return air has the same effect.

49. At time of fire in a metal mine lack of efficient ventilation system may be disastrous. For fire purposes each mine should have fans so placed as to be inaccessible to fire, have fire-proof housing, be capable of quickly reversing air currents if desired, and preferably should be placed on the surface. There should be a definite system of air splits such that the fire in one place may not necessarily fill the entire mine with poisonous fumes. There should be a system of doors near shafts in levels leading from shafts such that the shaft or any part of mine may be readily isolated in case of fire.

50. Recent experimental work in mines reveals that after smooth lining of shafts previously having ordinary exposed timbers there will generally be a reduction of friction to such an extent that additional air to the extent of 50 or more per cent can be handled by the same power. It is also found that in small fans with canvas tubing, the same amount of air will be delivered with 1/4 to 1/3 power saving by the use of fans with 1/2 to 2/3 the width of the standard fans. If the smooth lining is done by use of gunite, it also serves as fire proofing and for ordinary sized shafts say 6 ft. by 16 ft. it can be accomplished at cost of less than $10.00 per lineal ft. of shaft.

51. There has been in use in South Africa for several years an instrument known as the Kata thermometer, with intent to determine comfort or "cooling power" of air, taking into consideration temperature, humidity and air movement, but not considering effect of gases or dust. This instrument has been used to a limited extent by Bureau of Mines engineers and doctors in mines during the past few years, and while found unadapted for temperature much above 90 deg. fahr., considerable interesting data has been obtained and results will soon be published and additional work will be done.

52. While the cost of establishing a ventilation system for a large mine is variable, the cost of operation is not particularly burdensome, and operating cost will almost invariably be met by savings in compressed air, and in increased efficiency and health of employees, and frequently the savings effected cover the entire cost of the investment within a few years. A recent magazine article states that miners' tuberculosis costs the mining industry in the Rand, South Africa, upwards of $6,000,000 annually, and it probably costs the metal mines of the United States more than that sum, in addition to many lives and much misery, and practically all of this may be saved by adequate ventilation. By systematic work in dust prevention and ventilation betterments South African mines reduced the death rate from all diseases among the native miners of the Rand from 35 per thousand in 1910, to 13 per thousand in 1915, and to about 11 per thousand in 1920, the latter including the influenza epidemic. It is reasonable to believe that our metal mining companies will not be content to lag behind South Africa in progress and that ventilation betterments will take place rapidly.

PROBLEMS IN METAL MINE VENTILATION TO BE SOLVED

Herewith is a partial list of problems in connection with study of metal mine ventilation upon which work should yet be done:

1. The physiological effect of movement or velocity on saturated air at various temperatures should be studied, with the idea of determining what specific velocity would be the most effective at each temperature. Also a study should be made as to the effect of various velocities in very hot—say up to 130 deg. or over—unsaturated air with the idea of determining the exact velocity which would be most effective at each temperature and humidity.

2. The physiological effect of small quantities of mine air impurities, such as CO, CO_2, oxides of nitrogen, low oxygen content, etc., should be studied, in connection with varying temperatures, humidities and air movements. The writer is not inclined to agree with present-day idea that CO_2 is of comparatively little effect, as there appears to be a very decided physiological effect of very small quantities of CO_2 if the air is of high temperature and stagnant.

3. In view of the fact that movement of air is of vital importance, it appears desirable to study methods of moving or stirring air at working faces, such places at the present time usually having stagnant atmosphere. A study should be made of various makes and types of fans driven by water, compressed air, electricity, or otherwise.

4. A study should be made as to heat derived from fans, canvas or galvanized tubing, and other ventilating devices under various conditions, as to velocity, friction, etc.

5. A detailed study should be made as to the amount of heat given off, oxygen absorbed and CO_2 given off in the decay of timber. A partial study was made in the Bureau of Mines laboratory at Pittsburgh in 1914, and it was found that air in an enclosed cask with small quantity of moistened sawdust was quickly deprived of all of its oxygen. This study should be extended.

6. A detailed study should be made of methods, appliances, etc., for the cooling of underground air by water sprays, refrigeration or other method.

7. Detailed data should be obtained as to the applicability of the Kata thermometer to determination of air conditions in our mines. The Kata thermometer has been used to a slight extent by the Bureau of Mines in its ventilation studies, but not nearly to the amount that it has been used in South Africa.

8. While it has been thought that miners' consumption is due chiefly to siliceous dust, it is now believed that other causes are of almost equal importance, among them high temperatures, high humidities, lack of air movement, and possibly others. There is urgent need that further study be made to ascertain the exact relationship of each of these or all of them to the spread of miners' consumption.

9. While siliceous dust has been thought to be the main cause of miners' consumption, there appear to be conditions under which highly siliceous dust in large quantities is not harmful, and there also appears to be a probability that dusts other than siliceous may, under certain conditions, be harmful. A more detailed study should be made to ascertain the exact status of dusts of various kinds with relation to their effect on the health of those who breathe them.

10. In view of the fact that drilling in metal mines appears to be the chief cause of dust formation, there should be made an extended study of the use of wet stopers and other devices to prevent the formation of dust in mining.

11. A study should be made as to the effect of humidity on the human system, not only when temperatures are high, but also when they are comparatively low; also methods for the reduction or increase (if desired) of humidity should be studied.

12. A careful study should be made of the effect on dust content of air of mines by increasing or decreasing the relative humidity.

13. While the general impression is that coal dust is not harmful to the health of those who breathe it in the air of coal mines, there appears to be a possibility that coal miners ultimately are affected; and a study should be made as to dust in both bituminous and anthracite mines on the health of the miners.

14. Inasmuch as practically all explosives now used underground in metal mines give off considerable quantities of poisonous gases upon being fired, it is extremely desirable that a study be made with the idea of devising some explosive which will not generate dangerous gases.

15. In addition to the laboratory study of the effect of decay of timber, there should be made an extended underground study of methods to prevent timber decay with consequent formation of heat and CO_2 and financial loss to the mining companies.

16. Although the use of canvas tubing has greatly improved ventilation of dead-end working faces, the rapid decay of canvas tubing under certain conditions underground has prevented the introduction of this method of ventilation to a considerable extent. It is extremely desirable that a study be made to ascertain why and under what conditions canvas tubing decays rapidly underground, and to devise methods of treating the tubing to prevent such decay.

17. There appears to be an urgent necessity to educate metal mining men and managers as to the desirability or necessity for ventilation of all mines, and methods of doing this educational work should be devised.

18. Inasmuch as mining schools provide the greater portion of the mine managers, it is extremely desirable that more attention be given to the importance of ventilation in metal mines in the school courses, and methods for bringing about this situation should be studied.

19. Recently a study has been made in South Africa by Dr. Orenstein and Dr. Ireland as to the amount of physical work which can be obtained under certain conditions as to temperature and humidity. An extension of this study should be made to include not only temperatures and humidities, but also air movements and various amounts of air vitiation.

DISCUSSION

O. W. Armspach: The author has given some valuable figures concerning the effects of various temperatures and humidities on the body. I wonder if he has any definite figures as to just when the body will no longer function normally, that is, with respect to temperature and humidity. When will the heart beat be increased, when will the blood pressure, the respiration and temperature of the body be affected? At just what temperatures and humidities will these changes occur?

Author: Answering Mr. Armspach, I will say that Dr. Sayre, of the Bureau of Mines, and I spent two months, March and April, in the mines of Montana investigating that very subject, and we are just getting our tabulations out now. We found that when the temperature was above 80 deg. fahr. if a person tried to work even moderately, that his blood pressure fell 15 or 20 per cent in less than half an hour and that his pulse beat ran very high; my own ran as high as 180 from a normal of about 90. Another man who was a good deal stronger than I was had his pulse beat run from normal at 70 up to about 160. The body temperatures will run up to as high as 102.8 or even 103 deg. That is, working in still air at temperatures above 80 deg.

A Member: In speaking of a 80 deg. temperature do you mean wet-bulb temperature?

AUTHOR: Yes, that is, wet-bulb temperature above 80 deg. When you get up above 90 deg., especially when the air is saturated, it does not matter whether you are working or sitting still, you have extremely profuse perspiration. Your body temperature staying still will rise to 101, 102 or 102½ deg., and the pulse beat will run up to 150 or thereabouts. On the other hand, if that air is saturated and you subject it to a slight movement, say 200 to 500 ft. per min., you relieve yourself enormously of these symptoms up to the point where the air approaches the temperature of the blood. We found that when we sat still in 95 deg. saturated air, our temperatures, blood pressure and pulse beat acted as I have stated; our temperature and pulse beat went up high and our blood pressure dropped down, even sitting still, but just as soon as we moved that air, giving it a velocity of about 500 ft. per min., the conditions were almost comfortable. However, when the air approached about 98 deg. and was saturated we found that the more we moved the air, the worse the conditions were. When the temperature ran up to about 100 deg. (100 deg. saturated air was the highest we tried) we found it was almost unendurable if we moved the air at all. On the other hand, the moment we got into a place where the wet-bulb temperature was 102, the dry-bulb temperature 113 deg., comparatively dry air, we found that the conditions were not anywhere near as difficult to endure as where it was about 97 or 98 deg. saturated. When we left that air, wet-bulb 102 deg., dry-bulb 113 deg., we came out with our clothes perfectly dry and practically no perspiration. We found, however, that with that dry air of the above temperature, or saturated air of that temperature, if we moved the air fast, about 1000 ft. per min., it almost burned us up. On the other hand, if we moved the dry air about 150 to 200 ft. per min. it was perfectly comfortable. We have not our data complete yet but we expect to have it out in about two weeks.

E. S. HALLETT: Was the amount of oxygen reduced?

AUTHOR: In every case where we went underground we took air samples and in no case was the oxygen below 20.6 per cent—practically normal air.

O. W. ARMSPACH: One interesting point that ought to be pointed out here is the fact that Mr. Harrington states that when the temperature of the air was about 95 deg., increased air movement had no effect. That has been the theory that has been held for the last few years; that is, when the air is saturated and at practically body temperature it is no longer possible to remove heat from the body by convection, and instead of removing heat it carries heat to the body and the body will feel warmer as the air motion is increased.

AUTHOR: That is not quite true of unsaturated air, but of saturated air that is absolutely true. With the air not saturated you can get a very strong degree of comfort by a certain amount of movement.

O. W. ARMSPACH: By evaporation?

AUTHOR: Yes.

No. 606

A PLEA FOR BETTER AIR DISTRIBUTION IN VENTILATING SYSTEMS

By J. R. McColl, Detroit, Mich.

Member

FOR a great many years ventilating engineers have been designing systems on the basis of 30 cu. ft. of air per min. per individual, particularly in designs for school buildings. This seems to have originated in the idea that the carbon dioxide must be kept down to 10 parts in 10,000, on the old theory that ventilation was mainly a matter of maintaining a proper degree of purity of the air.

The old theories of ventilation have been supplanted by a belief that ventilation in schools, churches, theatres and the like, is mainly a matter of getting rid of body heat. Therefore if the air supplied in ventilation and the room itself could be entirely free from bacteria, dust and odors, the question of ventilation would be reduced practically to that of getting rid of body heat, assuming, of course, that the air is free from obnoxious, poisonous or other injurious gases not classified above.

It would be merely repeating what most ventilating engineers now know, to say here that the factors which concern the getting rid of body heat are air movement, temperature and humidity, all interdependent in their effect in the matter.

It is not the purpose of this paper to discuss the physiological effects of these three factors nor of the other mentioned factors in their relation to comfort. This has already been well covered before the Society in other papers, especially in those of Dr. E. V. Hill, whose synthetic air chart was accepted a year ago by the Society as the standard of measuring the degree of perfection in ventilation. It is rather my object to make a plea to ventilating engineers to give more study to the matter of air distribution—a subject which is especially vital in crowded rooms, such as school rooms, where individuals must sit for hours at a time and be kept comfortable and efficient.

Leaving out of consideration for the time being all effects in air movements caused by radiators or individuals, whose bodies act more or less as radiators, there are two forces only which will distribute air in class rooms, as schools are usually planned—namely, *gravity* and *projection.*

The first of these forces is utilized by all systems which affect both heating and ventilating by warm air. The old straight blast system is of this type. The air is delivered to the room at a much higher temperature

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than the normal room temperature, stratifies on the ceiling and settles as it cools. This settling is more rapid on the outside walls where the cooling is more rapid, but, nevertheless, there is more or less settling all over the room as shown by smoke tests which the writer has made. Systems using indirect radiation at the bases of flues, as in Chicago and Toledo schools would also come under this type of gravity distribution.

The other force of distribution is used in what I would designate as the projection systems where the air is projected or distributed by virtue of its velocity. The Beery system and the system designed by Dr. E. V. Hill, as published in the *School Board Journal,* are of this type. In years past, we have used the projection system more or less in the delivery of air and the Board of Education has come back to it in a more improved way in their latest Detroit schools.

For a good many years, the system which we have been using in Detroit schools has been a combination of the gravity and projection systems. Although the larger part of the heating has been done with direct radiation in what might be called a modified split system, a portion of the heating has been done with warm air in order to get the stratifying effect. The inlets have been placed sufficiently low and equipped with special diffusers so that some disturbance of the air at the breathing level is constantly induced by the velocities of the incoming air. The results from this system have been highly satisfactory, but the high cost of sheet-metal work for the individual ducts to each room have made a continuation of it prohibitive and we are now using the projection system, by simply using six special ceiling diffusers for each class room, and doing all the heating with direct radiation.

A school requires a fairly close balance in the distributing system to the various rooms. The only way we have been able to get this close balance is by the use of fairly large plenum chambers in which the velocity pressures from the fan are very largely converted into static pressure and from which the static pressure is converted with velocity pressures for the various ducts leading individually to the various rooms. We are able normally to get a system balanced within 5 per cent, and it will stay in balance even though several rooms are shut off. If a trunk duct system can be balanced within 25 per cent the engineer is lucky, but he's sure to be in bad luck if a few of the rooms happen to be shut off, for it will completely unbalance the whole system.

We have referred to air currents set up in the room by direct radiators and other sources, such as individuals, whose bodies act to a certain degree as radiators. When these are considered in connection with the currents set up by the rapid cooling of air along the outside walls, we have a very complicated problem in air movement with a more or less *churning* effect in any system. It is this fact, I think, which has saved the ordinary split system, which attempts to do all of the heating with direct radiators and all of the ventilating with air delivered to the room at the normal room temperature, from being a total failure. This air is frequently delivered through nothing but register faces in badly chosen locations rather than through well designed diffusers placed in well chosen positions, as, for example, the middle third of the wall opposite the greatest exposure of the room. When we consider that the heat transmitted through the walls to the outside and lost must come, for the most part, at least, in any system, from heat taken from warmed air coming in contact with those walls, we must admit that this and the currents already referred to as being set up by the radiators and individuals, will give a *churning* effect, which will assist in distributing the air over an occupied room. The degree of distribution, however, will be somewhat dependent upon the degree to which the radiators are working and the heat is being lost from the building. In other words, smoke tests which the writer has made in weather of varying degrees of severity seem to show a variation in this distribution which may be called more or less, accidental distribution.

My experience has been that a system which has been deliberately laid out as a gravity system or a projection system or a combination of the two, gives far better uniformity in distribution than systems which are more or less haphazard in their methods of control.

Now, that we have Dr. Hill's synthetic air chart as a means of determining the efficiency of ventilating systems, we predict it will not be very long before new values will be set on the question of proper volumes in ventilation. The old basis of 30 cu. ft. of air per pupil per min. will doubtless prove insufficient for some systems, and more than necessary for others, in order to obtain equal and satisfactory results. This will happen because one system distributes better than another and, without doubt, some of the old inefficient systems will be discarded.

The new Detroit systems have been designed for 30 cu. ft. of air per pupil, but it is expected to get along on 20 cu. ft. and possibly less. In the average school where the net class room floor area is about 50 per cent of the total floor area of the building, the ratio between the ventilating load and heat load is about 3 to 1. It is evident therefore, if the volume for ventilating can be reduced from 30 to 20 cu. ft. per min. per pupil, approximately 25 per cent of the coal bill will be saved.

I believe that ventilating engineers will eventually come to the recirculation of a large portion of the air in schools. This has already been tried with success and has resulted in marked fuel saving.

I also believe that something will be done in the near future on what might be called intermittent ventilation. Instead of maintaining continuous and uniform distribution, whatever the type, systems will be designed to give intermittent flow of air, making the maximum delivery velocities much higher than are now in vogue for continuous operation and tapering off to practically nothing before the next impulse. Dr. E. V. Hill, of Chicago, and W. F. Walker, of Detroit, as well as others have been giving this matter considerable study. The new Detroit systems are arranged so that it will be very easy to add this feature if the time ever comes for its application. The necessary intervals, of course, will be a matter of experiment and probably individual opinion.

DISCUSSION

J. D. Hoffman: I have no criticism on the paper proper, but I have one or two ideas to present for general consideration. These ideas have been called forth by certain references to the old and new theories of ventilation. Considering the second paragraph of Mr. McColl's paper, personally I still cling to the old school ideas, somewhat. As a Society, however, I know this is true of many members, we have been rapidly moving to the opposite extreme. We are talking away from the outside air idea and saying that all we need is to freshen up the air, temper it, moisten it, move it, and it does not make any difference whatever else happens to it, it is all right. I cannot bring myself to that belief. Maybe I am all wrong. I believe that this Society should talk more about the purity of the air. Ventilation is not simply air movement and air temperatures. I believe strongly in all that but there is more to it.

In quoting from the next to the last paragraph: "I believe that ventilating engineers will eventually come to the recirculation of a large portion of the air in schools." I think I can agree with that statement. In some cases, we could recirculate a considerable amount of air. This then becomes a matter of distribution as stated in Mr. McColl's paper. But until we have arrived at a real scientific method of distributing the air let us stick to the 30 cu. ft. of outside air as a safer proposition.

F. R. STILL: The subject of recirculation of air seems to hinge largely on the question of economy. If it comes to a choice between recirculation and the use of fresh air, it appears to me somewhat as to whether I wished clean water to wash in, or the water that some other fellow had washed in; even if you know that water has been put through violet rays, or some other purifier and there is no dirt discernible and you are positive that there are no germs in it, yet the fact that you knew that somebody had washed in it would make you hesitate to use it. I feel exactly the same about recirculating air in a building. There is nothing to recommend it except the saving in fuel.

In the Jackson School in Minneapolis two rooms were fitted up with the idea of carrying on experiments to determine the effect of better distribution of air. An air outlet nozzle was fitted to the corner of each desk. In this school there was an air washer with automatic humidity control and a wide range of speeds could be attained to regulate the volume of air discharged into the room (See TRANSACTIONS, Vol. 19, 1913, p. 328.) I want to call attention to this one development, however, which was that when only 10 cu. ft. of air per min. per pupil was delivered into the rooms the conditions were superior to those prevailing in other rooms in the same school where more than 30 cu. ft. of air per min. per pupil was being circulated. This was due to cleaner air, more desirable humidity and better distribution. Each pupil had his supply before it reached anyone else, or before it came in contact with anything in the room.

There is no question about distribution having a great deal to do with the satisfactory results obtained from a ventilating plant. The attempt now being made in the Detroit schools to get better distribution is very commendable and promises to be a very material advance in the art.

Smoke tests frequently disclose some very interesting things. We have recently seen some systems which would never be satisfactory if 60 or even 100 cu. ft. of air per min. per pupil was discharged into the room. With a good system of distribution it does not require so much volume, hence the best way to economize in fuel is to have a good system of distribution rather than to advocate recirculation.

E. S. HALLETT: I am very glad indeed to have Mr. McColl make this plea for better air distribution. It forms a very important part of the study in planning school building ventilation. It is not necessary to supply 30 cu. ft. per min. of air per person but it has been found to require about that amount to secure the proper air motion and to insure perfect distribution.

In the matter of the use of direct radiation in a school room most of you are aware that Mr. McColl and I do not agree and we may agree to disagree for the present. The important matter as I see it is that we shall have ventilation of some kind. We have heard one engineer after another stand up and say in this Society that the janitors were not operating the fans. We can not have distribution unless the fans are operated. I would build apparatus that compelled operation for heating that they may certainly have ventilation.

The matter of recirculation of the air is raised. It may seem repugnant to think of breathing air which has once passed through the building but what of the multitudes in rooms with radiators and no air change? To be sure the occupants come to like it. They would not exchange the radiator system for anything else. The odors are there but they are unaware of them. We must protect them by insisting upon adequate air distribution to every child. The re-use of air which has once been purified and now washed again is better than that which comes in fresh with a load of dust, smells, bacteria and smoke of a large city. We have many tests to show that this is true.

There are more important considerations than economy, as Mr. Still intimates, but in this instance economy is a determining factor in having ventilation at all. I want recirculation and repurification because it is effective and is so economical that everybody can afford to have it.

Now then as I see it, we come to this point, that we have got to have either all blast systems or we are going to have no ventilation. And I am going to fight to the last minute on having the blast system in order to get ventilation.

L. C. SOULE: I think this system of the ducts in the ceiling with the three diffusers is very ingenious. I would like to have Mr. McColl tell us a little more about the construction and size of the ducts leading from the plenum chamber to these ceiling ducts.

AUTHOR: It is only within the last year, that the Detroit School Board put a basement under their schools. Now it is possible to have a basement corridor under all first floor corridors, high enough to walk through, like that the Chicago and St. Louis schools have as a plenum chamber for tempered air. The corridors are capable of being washed down with a hose, kept entirely sanitary, and through these are carried the steam pipes, etc., for the direct radiation supply.

From these basement corridors or basement plenum chambers 10 by 20 in. risers go in the corridor walls to each room, the standard classroom being 23 by 27 ft., seating 40 pupils, two of these risers connect with horizontal ducts in the ceilings. The ceilings are constructed on the so-called pan type of construction, and two of those pans, one-fourth the distance from each end form the horizontal duct in the ceiling connected with those risers. In each of these horizontal ducts, which are also 10 by 20 in. are three diffusers, making a total of six in each classroom. They are about evenly spaced, so that the distribution is about the same in all directions. Recently we met and tried out three special diffusers in order to pick out the one that gave the best results. We put them up in a room of the same size in one of the old schools and made our smoke test. Two of the diffusers gave excellent results, one better than the other. We used various velocities, ranging from 50 or 60 to about 1500 ft. per min. We obtained the best results at from 90 to 100 ft. per min. This velocity spread the air evenly over the entire room. I do not care where the outlets are as long as the air can get out freely. We do not care about the direct radiation, except it shall be on the exposed side. You cannot blow a stream of air through a room and maintain a hole through that atmosphere. It will carry with it the surrounding air by induction. Therefore if you have direct radiation on the outside exposure the rising air will soon be caught in the downward current, and the impinging air will carry the air on the outside wall where it is to be used. Therefore we stipulate that the radiation shall be on the outside wall; a departure from the plan of having radiation on three sides of the room.

F. R. STILL: I want to correct what is apparently a misapprehension, judging from what Mr. Hallett said. My reference to fresh air had to do entirely with economy. The point I was trying to make is, if we can only determine how much air is necessary to give satisfactory, healthful conditions when we have a system that distributes the air thoroughly, we would certainly find that we require much less air than is our present standard and it is my thought that we can attain the same economy, by using less of it, than could be obtained by recirculating the air within the building, of which latter we would have to handle a very much larger volume in order to keep it up to proper sanitary standards.

J. H. DAVIS: Mr. Hallett spoke about the schools in St. Louis, in which the air was very satisfactory to those who were in them. I would like to ask what the record is of the majority of pupils as to health.

E. S. HALLETT: Offhand I couldn't give you the health record. My own opinion based on a comparison made this year by our Health Department is that it is not anything like as good. The doctor compiled statistics on two schools, one of them equipped as I would have them equipped, ventilated perfectly, one that I considered an ideally ventilated school, and the total number of cases of sickness in that school in which a child lost one or more days was 100 for the season; and the checked school nearest by, the same unit, the same kind of children, had 392 cases. There was no more epidemics in one than in the other; due, as I thoroughly believe, to the difference in the ventilation of the two schools.

L. C. SOULE: I think one of the big points in favor of Mr. McColl's system is that he gets rid of the long galvanized-iron ducts in the basement which collect dust quickly and which are unsanitary and objectionable for that reason.

No. 607

SOME DEVELOPMENTS IN CENTRIFUGAL FAN DESIGN

By F. W. Bailey[1], Buffalo, N. Y. (Non-Member)
and
A. A. Criqui, Buffalo, N. Y. (Member)

Up to about twenty years ago all centrifugal fans were of the straight radial blade type. Since that time the multiblade forward-curved blade fan, combining as it does higher efficiencies (in commercial sizes) and smaller space for a given capacity, with equal or lower first cost, has almost entirely supplanted the older type of fan.

In making this substitution, however, two very good characteristics of the old radial blade fans were lost. One of these was the pressure curve which rose continuously with a decrease in capacity and the other was the power curve which increased constantly with increasing capacity.

Fig. 1 gives a comparison of the pressure curves of these two types. Starting with maximum capacities of air, both curves show an increase of pressure with a decrease in capacity. The pressure continues to rise with the radial blade fan clear up to no load while with the forward curve blade fan the pressure stops rising when a certain point is reached and with further decrease in capacity shows instead a slight drop in pressure.

Fig. 2 gives a comparison of the power curves of the old fan with that of the forward curved blade fan. While with the radial blade fan the power increases constantly with the quantity of air, the power required for the forward curved blade fan increases at a much more rapid rate.

This combination, in the forward curved-blade fan, of the pressure curve which droops or even only flattens with decreasing capacity within the working range of the fan, with the power curve which has a more abrupt rise as the load increases, is sometimes the cause of serious trouble in fan installations. The flat portion of the pressure curve makes the fan very sensitive to resistance variations and if used at a capacity corresponding to this portion of the curve may make the fan run under or over the estimated capacity. This is particularly so if the friction of the system is slightly greater or less than was estimated or if an existing duct system is changed with a consequent change in resistance.

With a directly connected unit, when a fan which has an abruptly rising power curve, runs above the estimated capacity there is a decided danger of overloading the motor. With a motor driven forward curved

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blade fan run at this critical point in its range, it is necessary to supply an excess of motor power with the probability of running the motor at reduced capacity and therefore reduced efficiency, to guard against a possibility of the fan overloading. This represents a perpetual insurance payment to protect the motor when using a forward curved blade fan under these conditions. The old radial blade fan with its steep pressure curve was more self-adjusting to conditions and would vary but slightly from the desired capacity and power when the resistance proved different from that estimated.

TROUBLE IN DOUBLE WIDTH FANS

The flat or drooping portion of the pressure curve also causes trouble in double width fans or when two single fans blow into the same duct system, provided of course that the capacity comes within the range of this flat portion of the curve. If there is some difference in the resistances

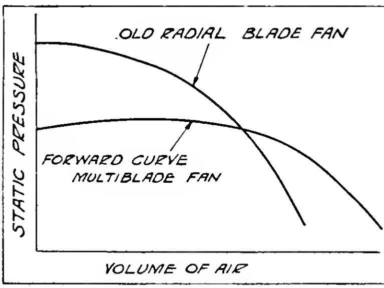

FIG. 1. COMPARISON OF PRESSURE CURVES OF THE RADIAL BLADE TYPE AND FORWARD-CURVE MULTIBLADE TYPE FANS

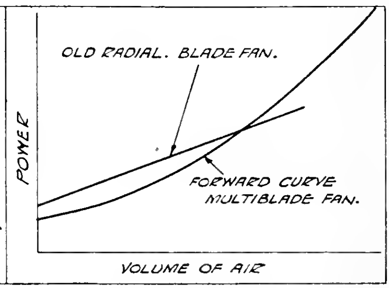

FIG. 2. COMPARISON OF POWER CURVES OF THE RADIAL BLADE TYPE AND FORWARD-CURVE MULTIBLADE TYPE FANS

at the two inlets, which frequently happens, the capacity of one wheel will be reduced without any corresponding increase in pressure. So this wheel has no power within itself to regain the load. Sometimes air will go in only one inlet, of a double width fan, and the other wheel will merely churn the air in and out its inlet. Of course, if the system is such that the fan runs at high rating far over on the pressure curve, a drop in the load on one wheel is accompanied by a distinct rise in pressure, allowing this wheel to regain its load. Here the load will be satisfactorily divided between the two wheels.

As will be shown later the forward curved blade fan is essentially a velocity fan. Air leaves the periphery of the wheel with very little static pressure but at a high velocity pressure. It is the function of the fan housing to transform the velocity pressure of this air into static pressure. Within limits the larger the scroll or cone effect of the housing, the greater the transformation and the higher the efficiency of the fan, but the greater the transformation the greater the hump in the pressure curve which means a more pronounced droop as the fan approaches no load. As we have seen a droop or even a flat spot in the pressure curve has decided disadvantages. The design of the housing of a forward curved blade fan is, therefore, a compromise between efficiency and reliability of operation.

The most desirable combination of fan characteristics would be to retain the good features of the old radial blade fan, that is the rising pressure curve from free delivery to no load, with a moderately increasing power curve, and at the same time have the high capacity range, insuring a small housing, together with the higher efficiency of the forward curved blade fan.

During the past few years a new type of multiblade fan has been developed which combines these good features. This new multiblade fan has blades which curve forward at the heel to meet the incoming air, but curve backward at the tip to discharge the air. Thus, while the blade of the forward curve fan is a portion of the surface of one cylinder or one cone, the blade of the new backward curved blade fan is formed from the surfaces of two cylinders or two cones as shown in Fig. 3.

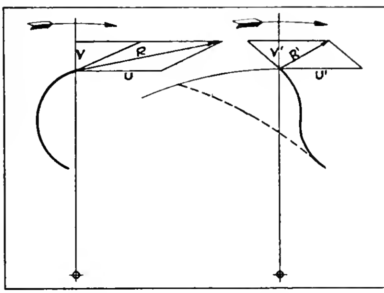

FIG. 3. COMPARISON OF THE FORWARD-CURVED AND THE BACKWARD-CURVED DESIGNS OF BLADES FOR CENTRIFUGAL FANS

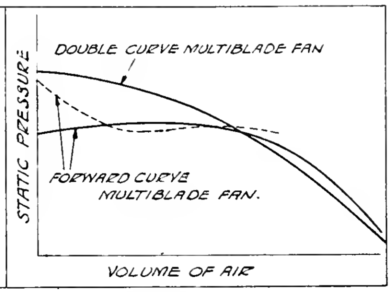

FIG. 4. COMPARISON OF PRESSURE-CURVES OF THE FORWARD- AND DOUBLE-CURVED-BLADE TYPES OF FANS

Before passing, it is well to remember that while multiblade fans are more efficient than the commercial types of radial blade fans, yet radial blade fans can be designed which are more efficient than commercial multiblade fans. To obtain high efficiency necessitates very long blades in proportion of the wheel diameter. This means a small inlet with small capacity and also a large diameter. A large diameter of wheel means high pressure or slow speed or both. Obviously the conditions which are necessary to produce high efficiencies in straight blade fans restrict their use to places where small quantities of high pressure air are required.

Fig. 4 shows the comparative pressure curves of a forward and a double curved blade fan. This double curved blade fan has the continuous rising pressure curve which when capacity decreases builds up a pressure to overcome the added resistance. With this double curved blade fan made double width, if, on account of the arrangement of the installation, one inlet of the fan is more restricted than the other, then the tendency will be for less air to flow through that inlet. But in such a case the restricted wheel will build up a pressure to meet the added resistance and thus retain its share of the load. A double width fan of this type will therefore divide the load between the two wheels throughout the whole range of capacities.

The comparative pressure curves shown in Fig. 4 represent two commercial designs of fans. Some standard forward curve blade fans show a greater drop in pressure at no load, while others, after flattening or drooping as capacity decreases, at extreme low capacities take a sudden rise in pressure. This sudden rise in pressure probably comes at too low a capacity to be of much practical value, particularly if preceded by a flat or drooping portion of curve. Other designs of double curve blade fans have a pressure curve which is flatter at no load but none show the flattening or drooping through the working range of capacity so common with forward curved blade fans. This difference in pressure curves therefore seems to be a characteristic difference between the two types of fans modified somewhat by individual design.

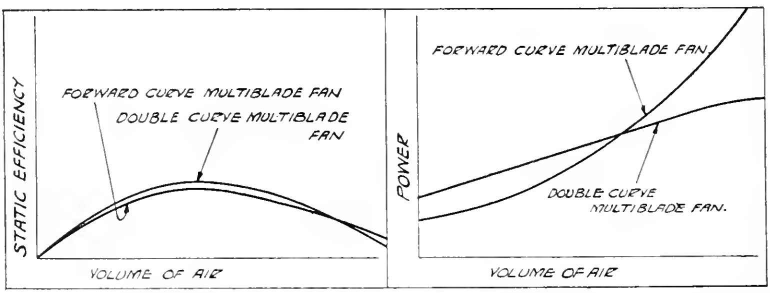

FIG. 5. COMPARISON OF THE STATIC EFFICIENCIES OF THE FORWARD- AND DOUBLE-CURVED-BLADE TYPES OF FANS

FIG. 6. COMPARISON OF POWER CURVES OF THE FORWARD- AND DOUBLE-CURVED-BLADE TYPES OF FANS

Fig. 5 shows the comparative static efficiencies of forward and double curved blade fans. Note that the efficiency for the double curved blade fan is slightly higher. This is a quite general characteristic as several types show this advantage when compared with various well known types of forward curved blade fans.

Fig. 6 shows a comparison of the power curve of the two types of fans. The fan with the double curved blade shows an actual dropping off in power at high capacities from the proportion of power to capacity shown through the earlier range of capacities. This combined with the continuous rising pressure curve means that the danger of overloaded motors is reduced. Many hundreds of these fans, direct connected to motors, are in actual operation and in no case has there been trouble from an overloaded motor due to the sensitiveness of the fan, although the ratio of the rated motor h.p. supplied, to the estimated power required, has in each case been smaller than would have been used had forward curved blade fans been furnished. This power curve on the double curve blade fan can be still further modified by special design so that the power will not increase at all above that required for the rated capacity, and with such a design it is absolutely impossible to overload the motor no matter under what conditions the fan is operated. However this design brings with it a reduction in the maximum capacity of the fan.

In one class of work the forward curved multiblade fan is supreme and will probably remain so. This is where very high outlet velocity is required, and noise is not objectionable. This fan is essentially a high capacity fan and may be used to great advantage for high outlet velocities.

The double curved multiblade fan is essentially a medium capacity fan and lends itself particularly to that class of ventilating work where very high outlet velocities are objectionable and yet it is decidedly a higher capacity fan than was the old radial blade fan. It has been developed in a number of different designs. Different angles of blades give fans with varying pressures for the same speed, or, what is the same thing, give varying speeds to produce the same pressure. They have been built with wheels having one inlet and the blades formed of the surfaces of two cones. They have also been built with a spider at the center of the wheel

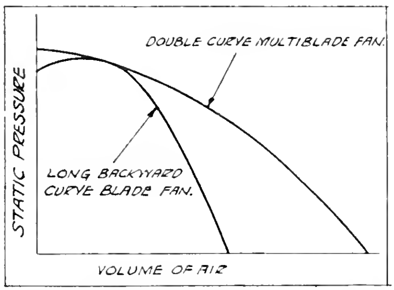

FIG. 7. CURVES SHOWING CAPACITY OF THE LONG BACKWARD-CURVE-BLADE FAN AND THE DOUBLE-CURVE MULTIBLADE FAN

and an inlet on each side in which case the blade is formed from the surface of two cylinders.

Double curve multiblade fans are inherently higher speed fans than are those with forward curved blades. With the double curved blade the speed can be varied through a wide range according to the design of the blade and the efficiency will be practically constant; but the lowest speed double curved blade fan will still be slightly higher in speed than the standard commercial type of forward curved blade fan.

Let us consider the parallelogram of velocities of the air leaving the blades of these two types of wheels as shown in Fig. 3. Assuming the same peripheral velocities U and U^1 and assuming the same radial velocities of air through the wheels V and V^1 (i. e., both fans running at the same point of capacity), the resultant velocity R of the air leaving the forward curved blade wheel will be much greater than the resultant velocity R^1 of the air leaving a double curved blade wheel. If in addition the two blades are of the same depth the static pressures developed in the wheels themselves due to centrifugal force will be the same. The forward curved blade, having much the higher velocity of air leaving, will also deliver the higher velocity pressure and though a portion of this will be lost in transformation by the fan scroll, still the total static pressure of the forward curved blade fan will be the higher. Hence in order to get an equal pressure from the double curve blade fan it is necessary to run it faster. The double curved blade wheel delivers a larger proportion of its pressure in the form of static pressure than does the forward curved blade wheel and therefore is not so dependent on the fan scroll to make the transformation from velocity to static. This accounts for its continuously rising pressure curve.

With the advent of the forward curved blade, trouble was experienced from noisy fans. The noise was attributed to the higher speeds at which these fans ran. Engineers and architects therefore got the habit of writing into their specifications clauses which limited the speeds of multiblade fans in order to insure large sizes or relatively slow speed makes. The noise in many of these cases was due to the high velocities of the air through the fans and piping systems. The high velocities were because the fans were operating near the critical points in their pressure curves and, the resistances of these systems having been overestimated, the quantities of air delivered were greatly in excess of the requirements. Specifying slower speed fans would hardly cure the trouble if the fans still ran at critical pressures and the resistances of the systems were still overestimated.

It must be understood that there are a great many factors which go to make up the noise in a fan unit, such as the arrangements of ducts, etc. but, in any case, noise is likely to occur when a fan runs considerably over the calculated capacity with the consequent higher velocities in the whole system.

It has been seen that the double curved blade fan (like the old radial blade type) is not so sensitive to resistance variation and will therefore run closer to the predetermined capacity when variations in resistance occur. But the double curved blade fan must be run at a somewhat higher speed. Specifications which fix speeds that practically prohibit the use of double curved blade fans would then seem to defeat their own purpose.

CONTINUOUS BACKWARD CURVED BLADE

The criticism may be made that a double-curved blade is not a theoretically correct form of blade; and that while the angles at the heel and at the tip of the blades may be correct, the intervening blade form does not give the proper acceleration to the air. It is probable that this form of blade would be unsatisfactory in a pump which handles a fluid 800 times as dense as air. With a fan however, this form of blade does give efficiencies which are very satisfactory. It is a question if a fan blade designed along theoretical lines would give any higher efficiencies and the double curved blade has the one great practical advantage of stiffness. A theoretically correct form of blade would probably be of continuous backward curvature and would reach much farther back on the periphery of the wheel in order to get the proper heel and tip angles. (See dotted lines Fig. 3.) This form however would be very weak as a beam in resisting centrifugal force and some form of intermediate flange might have to be used and the blades made each into two blades, placed end to end, in order to get a proper width of wheel. The reverse curve of the double curved blade adds a decided element of stiffness so that single blades, of the required length, are made to run at very high speeds without any danger of the blade collapsing from centrifugal force.

Another practical disadvantage of the long single backward curved blade is that the net air space between the blades is greatly reduced and the range of capacities is proportionately limited, which makes it necessary to use a larger unit for a given capacity than is required with the double curved blade type of fan.

This is illustrated in Fig. 7 which shows the capacity range of a fan of the long backward blade type in comparison with the capacity range of a double-curved blade fan, the diameter of blast wheels and speeds being the same. The long backward blade fan had by far the larger housing.

The double curve fan lends itself well to a variety of designs for various speeds and purposes and while it is perhaps too much to say that it will supplant the forward curved blade as completely as the forward curved blade fan has supplanted the old radial blade fan, yet it is safe to say that the double curved blade fan will find an ever increasing field in the handling of air for public buildings and for industrial purposes.

For a brief, non-technical history of the development of the centrifugal fan from its earliest inception see "The Centrifugal Fan" by the late Frank L. Busey (Transactions American Society of Heating and Ventilating Engineers, Vol. 22, 1915, p. 43).

DISCUSSION

F. R. Still: I do not think that the paper was written with the idea of deceiving anyone, but I do think that it is presented in a way which causes a mistaken impression being drawn from it.

All of you know that when the wind blows 25 miles an hr. you can hear it if there is any object in its path and your presence as an observer is one object which is bound to be in its path. Out on the plains in eastern Colorado where there is not a tree or a telegraph pole within one hundred miles one can hear the wind blowing at a velocity of 25 miles an hr. Such a velocity is about 2200 ft. per min. and therefore any time you move air at a velocity of about 2200 ft. per min. you are sure to hear it.

It does not necessarily follow that an outlet velocity of 2200 ft. per min. will be discernible in the upper rooms of a building, where blowers are usually installed, because the velocity is generally reduced before the air goes through the heating coils and is still further expanded to a lower velocity through the ducts, so that it ultimately enters the room at such a low velocity that it cannot be heard. All this, however, is a matter of engineering entirely apart from the blower itself.

All fans with curved blades having the concave side facing the direction of rotation will produce what we call a hump in the pressure curve. In other words, when the fan has a free discharge without any obstruction to the flow of air beyond the outlet of the fan, the minimum pressure is produced and the greatest velocity due to the flow of air results. As resistance is introduced which obstructs the flow of air, the fan running at a constant speed will set up a higher pressure and this continues up to a point somewhere between full open and all closed off where the pressure is maximum. After this point has been attained the pressure will make a very decided drop as more resistance is introduced and this will continue through quite a range until finally it begins to pick up again; when the blower is all closed off the final pressure will rise to a point about equivalent to the maximum which was previously obtained when there was less resistance to the flow. The location of the hump in the pressure curve can be changed to almost any point predetermined, from a resistance equivalent to 25 per cent of the full outlet area up to about 75 per cent of the outlet area, by changing the proportions of the fan housing, the inlet and the outlet.

A straight bladed fan of the paddle wheel type also produces a small hump, the peak of which is somewhere between 15 and 20 per cent ratio of opening. This latter type of fan, however, has no recovery of pressure after the maximum has once been reached, but the curve slopes downward as further resistance is added until it is all closed off, when it is very considerably below the maximum pressure at the peak of the hump.

Fans with blades curved backward, or with convex sides towards the direction of rotation and having a tangent at the tip of the blades considerably less than 45 deg. are entirely devoid of the hump but the maximum pressure attainable at the same tip speed is necessarily very much below what can be obtained with either straight bladed fans, or with forwardly curved blades. The result is that fans having blades curved backwardly have to run at such high velocities that they are very likely to be too noisy to be used in public buildings for heating and ventilating.

The point about the paper presented by the authors, which is not clearly brought out, is the fact that a fan having a wheel with blades curved forward should, when properly selected for the work it is to do, perform the duty just as perfectly as any other type of fan and will do it with higher efficiency, at slower tip speeds and occupy less space. Due to its slower revolutions there is less likelihood of mechanical noises due to vibration and as already related if the tip speed is slow one objection is entirely eliminated. Beyond question, if the inlet and outlet velocities are identically alike in both cases then there is no likelihood of any difference in the noise made by the two fans due to the air passing through them. Therefore, it seems perfectly logical that by eliminating as many causes for noise as possible less opportunity is offered for objectionable noises to result.

The fan referred to by the authors as originally designed had long triangular-shaped blades that terminated in rather a large sweeping curve near the tips of the blades. Due to the proportions of the housing and the inlet and outlet, the hump on the curve we have already referred to, occurred between 70 and 75 per cent orifice. The margin left to work on therefore lies somewhere between 75 and 100 per cent ratio of opening which necessarily made it a very difficult engineering proposition to select a fan that was not too large for the work it was to do. as in the latter case, if the resistance happened to be a little more than was calculated the pressure would slide over on to the wrong side of the hump resulting sometimes in very unsatisfactory operation and frequently set up a drumming sound that could not be eliminated. This drumming sound was due to pulsation of the air inside of the housing.

If an engineer's specifications happened to call for a given diameter of fan wheel it sometimes became almost impossible to select a standard fan of the type referred to which would not prove noisy. This objection was finally overcome by curving the tips of the fan blades backward, which has the effect of changing the tangent at which the air leaves the fan wheel, and instead of the direction of flow being directly forward, as it was before, it now leaves the wheel at an angle which is about 45 deg. less than it was before. The result of this is that the tips of the fan-blades have to run at a higher speed, and when plotted on a chart, the hump of the pressure curve has been moved from about 70 or 75 per cent over to about 50 or 55 per cent ratio of opening. This gives a wider range to work on with less liability of noise.

The application of fans or blowers, where the necessity for the modification of the blades herein referred to was first made necessary, was in connection with forced draft on automatic stokers, where it was necessary to set up pressures ranging from 5 to 7 in. and at the same time deliver very large volumes of air. The range through which such blowers have to work is anywhere from 20 per cent of full capacity up to 80 or 90 per cent. When the flow is restricted so that the slope of the pressure curve is downward from the hump toward the minimum capacity, the fans were so noisy and acted so peculiar that an objection was raised in some cases. It is quite evident that the fan blades when running at a fixed speed will fail to propel the air forward through a housing when a certain point of resistance has been reached, and as a result a multitude of eddy-currents are set up inside the housing. When this resistance is still further increased this internal condition grows worse until another point is reached where the pressure becomes so great that it overcomes the eddy-currents and then starts to build up the pressure again. The action of these eddy-currents has very much the same effect as pounding on the housing with a number of hammers would have, and in some cases the vibration is so pronounced that it seems actually dangerous to be near the fan. This can only occur, however, when running the fan at very high speeds and unusually high velocities; it never would be encountered in any heating and ventilating work. Noise, however, might be apparent under some circumstances although there would hardly be any apparent vibration due to the internal eddy-currents.

On general principles we are opposed to high-speed fans as they require extreme care in building, in setting up and in their operation. Any high-speed machine is very much more difficult to handle than a slow-speed machine and necessarily has shorter life. The only thing to recommend high-speed fans is the possibility of utilizing less space and perhaps the unit will cost less to purchase although it is bound to cost more to operate and will depreciate very much faster.

AUTHOR (A. A. Criqui): One of the aims in designing this fan was to eliminate a common trouble experienced with forward curved blade fans and to get one that would run more true to predetermined capacity and pressure conditions. Fig. 4 shows a pressure curve of the forward curved type. Suppose a fan is specified to give a certain amount of air at say 1 in. pressure. When that fan is put in operation on the job it may be found that there is 1½ in. resistance. If the fan is rated along the high part of the pressure curve, it may drop back in the hollow of the curve due to the higher resistance. This is the part of the curve which Mr. Still says will give it a drumming action. With this fan of the new type, having a continually rising pressure curve, there is no drop in the pressure curve at all and consequently there would not be any point where we would get any drumming action. In our old fan, having a drop in the curve, as shown above, if the resistance was not just as you figured it, the fan might operate back on the bad part of the curve, and you would get less air than specified, and possibly have to increase the fan speed. On the other hand, if your resistance was figured at 1 in. and you only had ½ in. resistance, then the air volume would increase with a rapid increase in power, as shown in Fig. 2, so that it would possibly overload the motor.

With a continually rising pressure curve of this kind, if you find a little more resistance than expected, the volume will drop down and the pressure run up rapidly to meet the condition, very near the point of the original estimate; while with a fan of the forward blade type that drops in pressure as shown in Fig. 4, as it drops back in capacity the pressure drops also, and together they will throw the capacity back much further on the curve. You would get probably 90 per cent of the air with the one fan and about 60 or 70 per cent with the other fan.

Now as far as noise is concerned and limiting the tip speed, we have installations of this fan running over 7000 ft. tip speed, quiet and absolutely satisfactory. I know that some architects try to limit the tip speed to somewhere around 3400 ft., and I think they base their opinion on the performance of forward curved fans, because forward curved fans at, for instance, 3400 ft. tip speed will give a pressure very much greater than a backward curved fan. When the tip speed is high with the double curve or backward curved fan it does not follow that your air velocity is high; it means a much lower air velocity than the same tip speed would produce with the forward curved blade fan. As Mr. Still says, out on the prairies he noticed that the wind could be heard blowing without any trees around; that shows it was air velocity itself and not any obstruction that caused the noise.

Regarding the pressure curve shown in Fig. 4 we found that having a fan that would approach the old steel plate fan characteristics so closely gave us very much less trouble on all installations than one where there was that drop or wave in the pressure curve.

I. B. COE: How does the double curve fan compare with the old steel-blade fan and its characteristics? That is, at a given velocity and given static pressure what would be the ratio of your peripheral speed to air pressure in the two cases? You know, of course, what the steel blade fans were. The static pressure was just about the peripheral speed. How does the double curved fan compare with that?

AUTHOR (A. A. Criqui): With a fan of the forward curve type the air pressure will be about twice the pressure corresponding to the peripheral speed. On the straight blade and the double-curved type the pressure will possibly correspond to tip speed. On the more pronounced backward-curve-blade fans the tip speed will be a little greater than it is on the straight-blade type, depending on design. We have one design of backward-curved blade in which the tip speed is not any greater than that of the straight-blade type. I may say also that in the old straight-blade type, the question of noise and tip speed was never raised, and those fans operated at around 4000 ft. tip velocity for an inch total pressure. Those systems ran probably from 1 in. to 1¼ in. static which would mean about 5000 ft. tip velocity.

I. B. Coe: In the old steel-plate fan I know the practical limits for quiet operation in places where quiet was necessary was from 3000 to 4000 ft. depending on the size of the fan.

Author (A. A. Criqui): I think there would be very few places you could put a fan in of the old type and keep it down to 3000 or 4000 ft. That would be 1-in. pressure. I would say most of your systems run about 1 in. static or over. We have a job now in which all the pressures are specified on supply fans at a 1½ in. static.

I. B. Coe: Mr. Still remarked that the hump in the curve could be carried to the left to any point. Has it been sufficiently brought out that this double-curved blade entirely removes the hump?

Author (A. A. Criqui): Yes, the pressure curve of the double-curved blade fan rises continually from free delivery to no capacity (see Fig. 4) and does not have a drop like that in the forward-curve-blade fan I might add that there was a recent test of noise which was made without our knowledge—at which a fan of this double-curved type, running at about 6300 ft. tip speed, was compared with the forward type at 4800 ft. speed, both fans running under the same conditions. A government inspector was given instructions to make a report on the noise. He reported the double-curve fan at 6300 ft. tip speed quieter than the other fan. Of course a fan of any design may make a noise in some systems, and I do not say that any particular design is noisy or quiet. It depends largely on the duct system and conditions of installation, the number of blades, etc. An expanding cone outlet connection will magnify any noise that you may have in the fan, and a good many other things enter into the proposition of noise.

C. A. Booth: I do not think anyone knows what is the limit of tip speed for noiseless operations, whether it is a multiblade fan or any other kind. A great deal more depends on the conditions of installation than on the tip speed, which is a factor of no practical importance, except as any increase in the tip speed means an increase of the air velocity throughout the whole system, and we know that somewhere we are going to reach the limiting air velocity for noiseless operation. This limiting velocity may be at the fan, in the ducts, or at the outlets. The conditions of installation, location of apparatus in respect to the rooms ventilated, the speed at which the air moves through the ducts and outlets, the construction and arrangement of the ducts, and the character of the foundation on which the apparatus is placed, are all factors. There are many conditions which affect the noiseless operation of ventilating fans.

I can recall, years ago, a very large installation which caused considerable expense to everybody involved, because the engineers insisted, and I would not question their judgment, that the vibrations of the motors were in synchronism with the vibration of the building, and thereby amplified the noise produced by the motors. As a matter of fact, I believed this was correct, or else that there was a megaphone effect such as we have sometimes found, so that the noise which was imperceptible in the main ducts, was disagreeable in the rooms. I have heard a great many arguments on the cause of noise with ventilating fans and how to remedy it, both among ourselves and in discussions of this kind. I do not believe that anybody can lay down limiting conditions, either as regards fan speed or air velocity in ducts, or anything of that kind, without being entirely at sea, unless the conditions of installation are taken into consideration.

A. G. SUTCLIFFE: I believe the point that Mr. Booth brings up and also the point that Mr. Criqui brought up in regard to tip speed limiting the noise is something that ought to receive consideration from all of the heating and ventilating engineers. It seems to me that heating and ventilating engineers have been educated by the older blower companies, perhaps, and to a certain extent by the older heating and ventilating engineers, to place in their own specifications too much stress on tip speed with the idea of limiting the noise; and as Mr. Criqui and Mr. Booth brought out, that is not necessarily the controlling feature. It seems as if perhaps that could be eliminated entirely in specifications and some other methods could be given. I am not just prepared to say what could be substituted for it, because in my experience I have never been able to find an instrument for measuring noise.

F. R. STILL: I am informed there is an instrument on the market for determining how much noise a machine makes. It is done by recording the vibrations and comparing them with the vibrations of some other familiar sound.

We all know that air traveling at a high velocity will make a noise, hence it is perfectly logical that an object traveling through still air at the same velocity will make a noise. It has been my observation that when no air is passing through a fan, as for instance, when the inlet or outlet is tightly closed, a fan wheel makes very little noise except such as is due to mechanical vibration; but just as soon as air is allowed to pass through it, when running at a high speed, the fan becomes noisy, the tone changing with the volume admitted.

Under normal operating conditions, as found in heating and ventilating plants in public buildings, the noise begins at about 3600 ft. velocity of the fan tips. When the speed is increased beyond 3600 ft. per min., which is about 41 miles per hr., the noise becomes more pronounced and if the speed is raised far enough nothing can be done to prevent it being distinctly heard all over the building.

I do not believe it is good engineering or sound advice to try to promote a high speed fan for public building work. I would rather have a fan with slow tip speed than one with high tip speed any time. One can never be sure as to whether or not a fan will make an objectionable noise when running at high tip speed. There is always some critical speed which varies in every installation, depending largely on the volume to be handled and the resistance to be overcome, whether the fan will make a noise. This is true of fans of every type and it is not due to the fan, the shape of its blades or the details of its construction. As a proof of this statement you can take a piece of sheet metal and by whirling the thin edge of it in the air at a high speed you can cause a noise that can be distinctly heard some considerable distance from it.

Author (A. A. Criqui): Regarding a 3600 ft. tip speed with a blade of the backward curved type, if you refer to the parallelogram of velocities, Fig. 3, in the paper you will see that the forward curve blade discharges the air at a much higher velocity than the backward curved blade does; so that this fan traveling at 3600 ft. tip speed would give you an actual air velocity that is very much higher than a blade of the backward curve type, traveling at the same tip speed. That means that this blade would possibly travel 50 per cent faster to give you the same air velocity.

Now if air velocity is the cause of noise, you can run this blade (the backward curved blade) very much faster and still not have any greater air velocity or noise. The angles of the blades on this fan are designed to pick up the air without shock and to let it go out without shock; while the forward-curved blade type picks it up without shock and sends it out at a high velocity. Consequently the air velocity leaving the forward curve type is very much higher, and for that reason you cannot take these two types of blades and compare them on the same standard and say that a certain figure is the limit for tip speed.

Author (F. W. Bailey): We do not try to show in this paper that tip speed in itself is an advantage, or that the higher the rated speed of a fan the better the fan, but we do want to show that the great advantage is in the sloping pressure curve and to get that advantage, the advantage of the sloping pressure curve of the double-curve-blade fan, it is necessary, apparently, to have a higher tip speed than with a forward curve blade fan.

F. R. Still: Between 50 per cent ratio of opening and all closed off represents a condition in which we would not select a fan for a given duty. It would be foolish for anyone to attempt it. The fan would be all out of proportion as it would be too large for the work it is expected to do. A fan should always be selected which will have a ratio of opening between 50 and 100 per cent.

I maintain that consideration of the tip speed of a fan is quite as important as inlet and outlet velocities. In fact, one is likely to get into greater difficulty by permitting excessive tip speed than by excessive outlet velocity because provision can be made to correct the latter to a large extent, but it would be extremely difficult, if at all possible, to correct the former.

Author (A. A. Criqui): Regarding Mr. Still's last remark, I wholly agree with him that a fan should not be selected that will run back in the hollow part of the pressure curve. (See Fig. 4.) But the trouble lies in the fact that specifications often call for a certain pressure condition, for which the fan is selected on the high part of the pressure curve. And when this fan is put into operation on the job it is found to be operating under a different pressure than that specified, which drops the operating point back to the hollow part of the curve and results in a lower capacity than that specified.

This is where you run into trouble, while if you had used the type of fan having a continually rising pressure curve, you would not have run into that difficulty.

No. 608

REPORT OF THE COMMITTEE ON RESEARCH

To the Members of the Society:

Since the Committee's report presented at the Annual Meeting of the Society in Philadelphia last January, the work of the Research Laboratory has continued along the lines outlined therein and very satisfactory progress can now be reported.

At a meeting of the Committee on Research, held in Philadelphia on the closing day of the Annual Meeting, the program of the Committee's work was extended to cover several additional subjects and the Acting-Director was authorized to increase the personnel of the Research Laboratory staff. In addition, much of the extension work at the various universities was approved and authorized for continuation in cooperation with the Research Laboratory. The work of the Laboratory now comprises the following subjects:

RESEARCH WORK AT THE PITTSBURGH LABORATORY

1. Infiltration.
2. Heat loss through building material.
3. Standardization of dust measurements.
4. Study of combustion in domestic heating boilers.
5. Checking application of heating boiler testing code.
6. Temperature, humidity, and air motion effects on health.
7. Study of ozone concentrations.

RESEARCH WORK AT UNIVERSITY LABORATORIES AND ELSEWHERE

Heat Loss through Building Materials:
Armour Institute of Technology. Morgan Park Company.
Pennsylvania State College. University of Minnesota

Radiant and Convected Heat:
University of Minnesota. Yale University.

Heat Losses from Pipes Buried Underground:
University of Michigan.

Study of Pipe Sizes:
New York University. University of Texas.

Air Friction in Pipes as Affected by Density:
University of Pennsylvania.

Chimney Investigations:
Carnegie Institute of Technology.

Ventilation Problems:
Minneapolis School Board. St. Louis School Board.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, June, 1921.

AIR INFILTRATION:
McGill University.

HEAT LOSS THROUGH GLASS:
Rensselaer Polytechnic Institute.

COMPARATIVE ECONOMY IN HEATING BY DIRECT RADIATION AND BY AIR HEATED WITH STEAM COILS:
Ohio Northern University.

During the early months of the year, Acting-Director Scipio travelled widely amongst the various universities and industrial research institutions and has formed, for the Laboratory, many valuable affiliations. As a result of this, the work of the Laboratory has been very liberally extended, much to the advantage of the research project. The Acting-Director has also been able to give much careful attention to the work now in progress at the Pittsburgh Laboratory, with the result that the installation of the cold-room equipment and the other research investigations being carried on there, have been materially advanced.

Some of the results of the recent work of the Laboratory are being presented at this session of the Semi-Annual Meeting and will place on record much additional data of value to heating and ventilating engineers. In addition, certain research papers which have been contributed at the request of the Committee on Research will be here presented and discussed for the benefit which they may bring to the research work. The value of this data as placed on record in the Society's proceedings cannot be overestimated.

At the present time the expenditures from the Research Fund are somewhat greater than the receipts, and the cash balance of the Research Fund has decreased from $17,131.56 on July 31st, 1920, to $13,488.32 on May 31st, 1921. However, the subscription invoices for 1921 have, under instructions from the Committee on Research, just recently been sent out and it is the expectation of the Committee that the receipts during the remainder of the year will again rise in excess of the expenditures.

The drive for funds which the Committee on Research had intended to carry out in the early months of the year, has been postponed owing to the present financial conditions and will, undoubtedly, now be withheld until the opening of the Chapter season in the early Fall. It is the belief of the Committee that financial conditions will then be so much improved that the drive may be more successfully carried out, and that the industrial conditions will influence everyone in heating and ventilating work to strongly support this great movement, so that the scope of the Research Laboratory's work may be materially extended.

The Committee on Research again reiterates its appreciation of the generous and whole-hearted support of the entire engineering profession in connection with this important work. The success of this research movement is dependent entirely on this cooperation and it is a matter of great satisfaction to not only the Committee on Research but also to the Officers of the Society, that, at the close of its second year of actual operation, the Research Laboratory of the Society has not only been able to hold its own in carrying out so ambitious a program, but has even been able to far exceed the expected range of subjects for research.

JAY R. MCCOLL, *Chairman.*

No. 609

VENTILATION TESTS OF SCHOOLROOMS AT MINNEAPOLIS

By L. A. Scipio, Pittsburgh, Pa.
Member

During the past year, considerable interest has been manifested in the ventilation tests being carried out in one of the public schools of Minneapolis, Minn., which were initiated by George F. Womrath, business superintendent of the Minneapolis Board of Education, and his corps of engineers in cooperation with the Research Laboratory of the Society. This work was to have been started last December but owing to weather conditions and the delay in completing the necessary arrangements, it was not actually begun until the middle of February.

The building in which the work is done is known as the Whitney School. This is a one-story frame building with brick veneer and contains six class rooms, an administration suite, a gymnasium, and a boiler room. The basement is entirely excavated and the attic space is some 5 ft. high so that the building offers an unusual opportunity for installing and making a study of the various types of ventilating systems.

On being notified that all was in readiness, O. W. Armspach of the Research Laboratory proceeded to Minneapolis in order that he might observe and be of any assistance in explaining the method of applying the results of the tests to the synthetic air chart as a standard.

The building was originally equipped with a system known as the Minneapolis individual unit ventilating sets, separate units being furnished for each class room, the gymnasium, and the office suite. In addition to these units, three of the rooms were fitted with different types of ventilating systems—namely, the Wheeler open window system, the Beery system, and the Moline univent system. Also, one of the Minneapolis units was fitted with a St. Louis ozonator.

The Minneapolis individual ventilating unit consists of the following apparatus: A fan having a capacity of 1500 cu. ft. of air per min., driven by a ¼ h.p. direct current motor with a rheostat speed control-a tempering coil, a re-heating coil, and an air washer of the spray nozzle type. There is also a water re-circulating system consisting of a small motor-driven rotary pump which furnishes the pressure for the spray nozzles as well. All of this apparatus is enclosed in a galvanized iron casing and the air which comes from the outside, after being drawn through the casing, is forced into the room through a galvanized duct which terminates in an adjustable diffuser located in one end of the

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, June, 1921.

room. Each room is supplied with a vent duct in the cloak room adjoining, and the exhaust air is discharged through a roof ventilator. Arrangement is made by means of by-passes and dampers for re-circulating the air.

All of the individual units are on a Bishop-Babcock temperature control system which operates as follows: The fresh air coming from the outside is regulated by means of a fresh air damper, automatically controlled by a thermostat and adjusted to regulate the temperature of the air entering the tempering coil to about 42 deg. fahr. The damper controlling the incoming fresh air from out-of-doors is operated from the same air line which operates the re-circulating damper so that the temperature of the air supplied to the ventilating unit is controlled at a fixed point. Arrangement is made whereby the air entering the washer is about 46 deg. fahr. There is no humidity control covering the temperature of the water supplied to the air washer. There is a thermostat in the fan discharge controlling the steam supplied to the re-heating coil and this thermostat is set at 68 deg. fahr. The above temperatures furnish a humidity in the room of approximately 45 per cent. The engineer states that a larger amount gave trouble because of water forming on the windows of the class rooms when the outside temperature was 10 deg. or less above zero.

OPEN WINDOW SYSTEM

One of the rooms was provided with an open window system installed in accordance with directions provided by Mr. Wheeler of Bridgeport, Conn., who came in person to Minneapolis and explained the method of installation. The amount of direct radiation in this room was increased to 320 sq. ft. and was so installed as to have the radiators extend under the full width of the windows where they were placed. This system provides for the air coming from out-of-doors by opening the windows to the required degree. The children are protected from the draft coming directly upon them by means of a glass shield 12 in. high placed in front of the opening. The air leaves the room through two ducts in the ceiling on the opposite side of the room from the windows and discharges through the roof by means of two 20-in. ventilators.

Another room is fitted with the Beery system, the distinguishing characteristics of which are, that instead of the air from the fan being discharged through the regular opening at the end of the room, it is led through a duct in the attic to two diffusers placed in the center of the ceiling. These inlets are of special construction and are provided by the manufacturers of the Beery system. The outlets, four in number, are located in each corner of the room and are joined together under the floor where the discharge is led to the vent flue which extends through the attic and to the outside. It is possible to recirculate with the system as well as with the Minneapolis ventilating unit. In the use of the Beery system, the radiation consists of 300 sq. ft. of pipe coil surface suspended from the ceiling around the walls of the room.

Another room was equipped with a Moline univent unit. The air supplied for the unit is obtained by raising the window and drawing the outside air through a close fitting sheet metal connection which extends to the sash line. As the air leaves the unit, it is blown in a vertical direction against the ceiling which causes the diffusion, and it then leaves the room through the regular vent opening provided by the Minneapolis system.

The room which was equipped with the ozonator required no modification. The ozone unit is placed in the ventilating unit proper and the air passes over it as it enters from the outside.

Since but one test was made at this time, it could be considered only of a preliminary character so that nothing definite can be said about the results. It was hoped that it would be possible to continue these tests

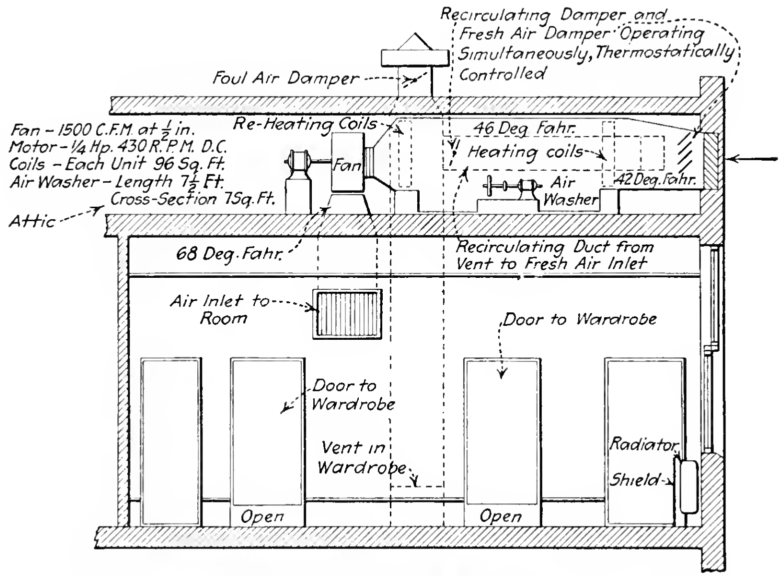

FIG. 1. DIAGRAMMATIC VIEW OF THE MINNEAPOLIS INDIVIDUAL-UNIT VENTILATING EQUIPMENT

regularly throughout the remainder of the year but suitable instruments were not available for performing the work. Orders were at once placed for other apparatus, but as this is of rather a special character, considerable time was required before it was ready for delivery. It finally arrived the latter part of April, and in order to make a further preliminary study, we returned to Minneapolis to assist in getting the work under way and to stimulate interest in it. On this visit, ten tests were carried out in order to familiarize the operators with the use of the instruments and to discover and point out any minor defects in the systems in order that they might be remedied before starting the work on a more extensive scale so that in the future it could be carried on with facility. These tests covered a period of more than a week, but the outside temperature conditions were about the same for all of them and in order to make a study which will be sufficiently thorough from which to draw conclusions, it would be necessary that the tests be carried out over a considerable period and a range of seasonal conditions.

Arrangements have been made whereby the work will be resumed next Fall and two or three complete tests each week will be run throughout the year. It is felt that this will then give us sufficient data on which to base some more or less definite conclusions and will contribute a large amount of valuable information which may serve as a guide for improving ventilating conditions in general. It should be borne in mind, however, that perhaps none of the various systems as embodied in this school are in their perfected conditions, and this is especially true of the regular mechanical system which is without automatic humidity control, proper distribution or arrangement of ducts and registers. It is hoped that we will be able to secure enough interest in the subject in some of the other cities of the country to carry out a similar kind of work in different locations.

The Minneapolis Board of Education deserves a great deal of credit for being the first to initiate a piece of work of so much importance and which promises to be of such great value to the country at large.

DISCUSSION

O. W. ARMSPACH: I might add that the Research Laboratory also assisted in tests made in St. Louis schools, especially the Bryan-Mullanphy School, which was equipped by Mr. Hallett. This school is equipped with ozonating units, and three tests were conducted, one with the ozone unit in operation, one without the ozone units in operation, and another with the ozone units in operation with 64 per cent of the air recirculated.

These tests, as was said, are of a preliminary nature, and the reason for the Research Laboratory's assistance in the tests is that they give direct information concerning the application of the synthetic air chart to such work. It is of prime importance at this time to determine the correct use of the air chart and to study the application of the instruments necessary to obtain the data.

These are not the official or final tests of the Research Laboratory in any way, but they are extremely valuable in that they point out the imperative changes to be made in both test instruments and mechanical equipment, and the information obtained is being directly applied to more detailed and extensive plans for experiments to be carried out by the Research Laboratory.

No. 610

APPARATUS FOR TESTING INSULATING MATERIALS

By F. B. Rowley, Minneapolis, Minn.
Member

THE subject of heat losses through various building materials is one which has attracted the attention of a great many experimenters and one in which investigations have produced widely-varying results. This wide variation in results may be accounted for partially by the different methods which were used by the various experi-

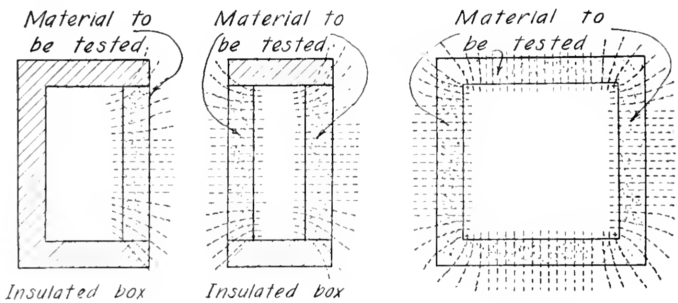

FIG. 1 FIG. 2 FIG. 3

VARIOUS ADAPTATIONS OF THE HOT-BOX METHOD OF TESTING HEAT TRANSMISSION

menters and partially by certain factors which are very difficult to determine accurately or to keep constant throughout the tests.

There are two general methods in use for this work, first, that known as the hot-box method, and second, that known as the hot-plate method. The essential difference between the two is that in the former the heat is conducted to and from the insulating material by convection currents of air, while in the latter it is conducted to and from the material by contact with metal plates. The results obtained by the two methods will differ according to the surface resistance of the material under test.

The apparatus used for the hot-box method has taken two general forms: First, that in which a well-insulated box is constructed with an open end, and the material to be tested is placed in the open end of the

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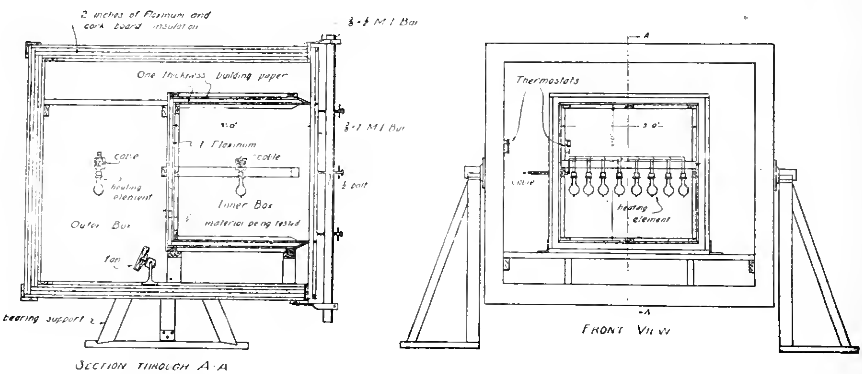
box; the temperature inside the box is maintained at a higher point than that outside and the heat supplied to the box is measured; this amount minus the constant loss from the box is taken as the heat transmitted through the material under test. In this method the box must be previously calibrated to determine the constant for the box.

There is an uncertain loss in this method known as the *end loss* through the material. Fig. 1 represents a cross sectional view of such a box with the material in place, which is in use at the University of Minnesota. The path of the heat travel through the insulating material is shown by the dotted lines passing from the inside to the outside. It is evident that some of these paths of heat lead from the inside surface directly out through the ends of the material under test and do not pass through in



FIGS. 4 AND 5. SECTIONAL AND FRONT VIEWS OF THE DOUBLE BOX IN USE AT THE UNIVERSITY OF MINNESOTA FOR TESTING HEAT TRANSMISSION

a line normal to the surfaces. This is an uncertainty which will affect the results a greater or less amount, depending upon the thickness of the material and the size of the specimen tested.

This method has been somewhat modified by using a box with two open ends, as shown in Fig. 2. This does not eliminate the so-called *end loss,* but minimizes any error which may result from calibration of the box proper, as there is less surface to the box compared to that of the insulating material.

The second generally accepted method is that of constructing a cube of the material to be tested, as shown in Fig. 3. This method eliminates the error which might arise from the uncertainty in calibrating the box, but on the other hand, introduces an uncertainty as to the path of travel of heat through the corner of the cube. It is generally assumed that the average between the inner and outer surfaces represents the surface exposed to heat. This is, however, very uncertain, as for instance, if we assume a 2 ft. cube of the material and assume the material to be 4 in. thick. The outside surface would be 24 sq. ft. and the inside surface would be 10 2/3 sq. ft. It is very doubtful whether the average of these two would give the true results as to the actual exposed surface. It was in order to eliminate the uncertainties due to end losses and box calibration that the double box was designed and built.

Fig. 4 represents a front view of this box with the insulating material removed, and Fig. 5, a vertical cross-section through the line AA. As shown by Fig. 5, the box is constructed with a double wall, or really it is one box placed within another. The open faces of both boxes are brought to the same plane and the insulating material to be tested covers both boxes. The temperature is maintained uniform in both the outer and inner boxes. It is evident that with the same temperature in both boxes, there should be no heat travel from the inner to the outer box, or vice

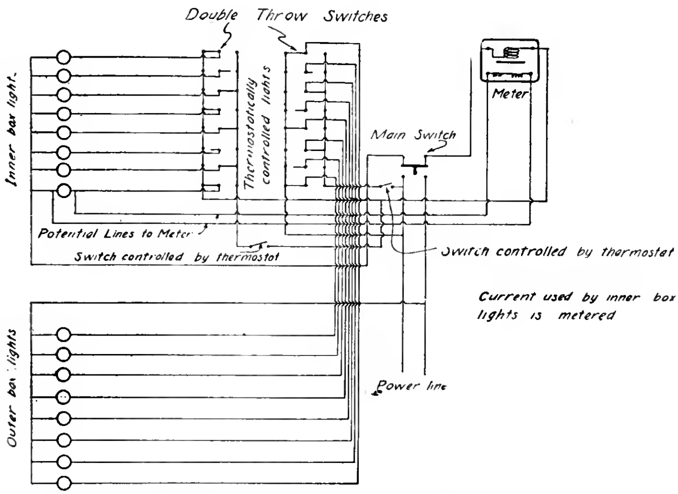


FIG. 6. WIRING DIAGRAM FOR THE HEATING ELEMENTS IN THE DOUBLE BOX APPARATUS FOR HEAT TRANSMISSION TESTS

versa, and that the only loss of heat from the inner box will be out through the insulating material covering the open face.

The flow of heat through the insulating material will be normal to the surface at all points. There will be no tendency for the heat to pass out diagonally at the edges of the inner box as this outer or guard ring will be supplied by heat from the outer box. The heat supplied to the inner box is measured and the only loss is that passing out through a definite area of the insulating material.

The heating elements used in this work have been electric bulbs surrounded by heavy opaque paper guards not shown in the drawing. One set of heating elements supply heat to the inner box and the second set supply heat to the outer box. The heat is metered to the inner box only, as it is immaterial how much heat is used in the outer box, so long as the temperature is kept constant and equal to that of the inner box. The heat is controlled by means of electric thermostats loaned to the Laboratory by the Minneapolis Heat Regulator Company, and it has been found possible to obtain a very close regulation.

In order to avoid unequal distribution of heat around the inner box a small disc fan was placed in the box as shown in Fig. 5. This fan circulated the air and maintained a uniform temperature throughout the space. No means of circulation was used in the inner box. Both boxes were well insulated as shown. The object of insulation was to avoid error from slight variations in temperatures which might occur throughout the test.

The outer box was first constructed and used as a single box for testing materials and was later converted into the double box as shown. This accounts for the fact that the size is somewhat larger than would be necessary to inclose the 3 ft. inner box.

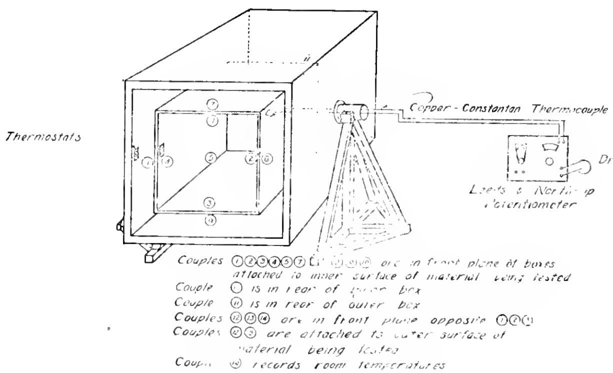


FIG. 7. OBLIQUE VIEW OF THE DOUBLE BOX SHOWING LOCATIONS OF THE THERMO-COUPLES

The complete unit is mounted on trunnions so that the material may be tested in a vertical or horizontal position. The details of this construction, the locations of heating elements, thermostats, etc., are well shown in Figs. 4 and 5.

WIRING DETAILS

Fig. 6 shows the wiring diagram for the heating elements. As is shown, each set of elements is under the control of a thermostatic switch. Each heating element is connected by a single-pole double-throw switch so that the load may be either carried on a thermostat or as a constant load. The object of this arrangement is to permit as much of the load as possible to be carried constant with only a small portion controlled by the thermostat. The current to the inner box was metered by a Westinghouse watt-hour meter.

Fig. 7 is an oblique view of the box, showing the location of the thermocouples. Temperatures were taken with a Leeds & Northrup potentiometer using copper constant thermo-couples.

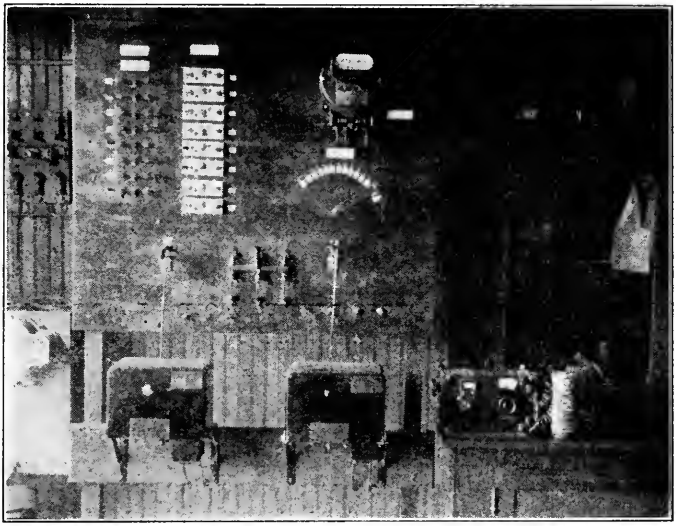


FIG. 8. SWITCHBOARD USED FOR CONTROL OF HEATING ELEMENTS IN THE DOUBLE BOX

The arrangement of the instruments on the switchboard is shown in Fig. 8. This apparatus has been in use for the past year at the University of Minnesota and has given very uniform results, as shown by the following data of the typical tests on the two thicknesses of 1 in. insulating material.

TABLE 1. AVERAGE RESULTS OF HEAT TRANSMISSION OBTAINED WITH A WELL-KNOWN COMMERCIAL INSULATING MATERIAL, EXPRESSED AS CONSTANT K.

Thickness of material, in.	Number of Layers. (No air space between layers)	K.
1	1	0.325
1	2	0.17
1	3	0.13
1	4	0.105
½	1	0.52
½	2	0.285
½	3	0.215
½	4	0.16
	(½" air space between layers)	
½	2	0.235
½	3	0.16
½	4	0.125

TABLE 2. RESULTS OF TEST OF TWO 1-IN. THICKNESSES OF INSULATING MATERIAL
(RUN No. 18, March 10, 1921)

Time	Meter	Thermo-Couple Readings, deg. Inner surface of material Inner Box					Outer Box							Outside Surface			Room
		1	2	3	4	Aver.	1" from sur. 5	Rear 6	7	8	9	10	11	12	13	Aver.	14
12:45	890.6	148	147	147	148	147.5	151.5	152.5	149	149	147	152	153.5	61.5	62.5	62	59
1:00	892.0	148	147	147	148	147.5	151	152	149	148	147	152	152.5	61	62	61.5	59
1:15	893.3	148	147	147	148	147.5	151.5	152.5	148	148	147	151.5	153	61	62	61.5	59
1:30	894.6	148	147	147	148	147.5	151	152	148.5	148	146.5	151.5	152	61	62	61.5	59
1:45	896.2	149	147	147	148	147.8	151	152.5	149	148	147	151	151.5	61	62	61.5	59
2:00	897.7	148	147	147	148	147.5	151	152.5	148	147	146.5	151.5	152	61	62	61.5	59
2:15	899.2	148	147	147	148	147.5	151	152.	148.5	147	146	151	152	61	62	61.5	59
2:30	9.008	148	148	147	148	147.8	152	153	148	147	146.5	151	152	61.5	62.5	62.0	60
2:45	902.1	148	147	147	148	147.5	151.5	152.5	148	147	146	150.5	151.5	61.5	62.5	62.0	59
3:00	903.8	148	147	147	148	147.5	151	152	147.5	147	145.5	150.5	151.5	61.5	62.0	61.8	60
3:15	905.5	148	147	147	148	147.5	151	152	147.5	147	146	150.5	151.5	61.5	62.5	62.0	59.5
3:30	907.1	148	148	147	148	147.8	151.5	152	149	148	147	151	152	61.5	62.5	62.0	60
3:45	908.8	148	147	147	148	147.5	151	152	147.5	147	146	150.5	151.5	61.5	62	61.8	60
4:00	910.5	148	147	147	148	147.5	151.5	152.5	147.5	147	146	151	151.5	61.5	62	61.8	60
4:15	912.0	148	147	147	148	147.5	151.5	153	147	147	146	150.5	151.5	61.5	62.5	62.0	60
4:30	913.6	148	147	147	148	147.5	151.5	153	147.5	147	146.5	150.5	151.5	61.5	62.5	62.0	60
4:45	915.2	148	147	147	148	147.5	151.5	153	147	147	146	150.5	151.5	61.5	62.5	62.0	60

Readings 1, 2, 3, and 4 are on inside surface of insulating material of inner box.
No. 5 is 1 in. from front surface of inner box and No. 6 is about 6 in. from rear end of inner box.
Nos. 7, 8, and 9 are readings 1 in. from surface of insulating material in outside box.
No. 10 is in outside box near thermo-couples.
No. 11 is in rear of outside box.
Nos. 12 and 13 are outside surface temperatures, front of inner box.
No. 14 is room temperature 4 to 5 in. from material over inner box

No. 611

A NEW THERMAL TESTING PLATE FOR CONDUCTION AND SURFACE TRANSMISSION

By F. C. Houghten[1], Pittsburgh, Pa. (Member)
and
A. J. Wood[2], State College, Pa. (Non-Member)

THIS paper summarizes the investigations of the past six months as conducted jointly by the Engineering Experiment Station of the Pennsylvania State College and the Research Laboratory of the American Society of Heating and Ventilating Engineers.

In the early study of the problems by the Director of the Research Laboratory and the Staff of the Engineering Experiment Station, Pennsylvania State College, careful consideration was given to the methods of obtaining constants for the various materials used in building construction. The tests in which the Research Committee of the Society were concerned, fitted into a general program followed for some years by the Engineering Experiment Station. The special facilities available at the Pennsylvania State College, led to the present cooperative plan as arranged by the late Dean Allen of the Laboratory and Dean Sackett of the Station.

Among the preliminary designs, was one for the construction of a guarded box built on the same general principle as the large animal calorimeter at the College which is recognized as a successful type of a large guarded box. When these plans were completed, it was found that no less than 16 heating elements would be required in its construction and there would be about 45 temperatures to be observed and recorded. Because of the inherent complications both in the construction and operation of the guarded box, this method was abandoned.

The possibility of adapting the well-known plate method to conditions for obtaining both conduction and surface transmission, led to the construction of a large air-cooled plate from which constants have been determined for corkboard for temperature differences varying from 50 deg. to 70 deg. fahr. Corkboard was selected because of the definite constants determined by different investigators. A comparison is made of the results from this plate with those obtained by using the hot box and the water cooled plate.

The Thermal Testing Plant of the Pennsylvania State College con-

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[1] Representing A. S. H. & V. E. Research Laboratory.
[2] Professor of Railroad Mechanical Engineering, The Pennsylvania State College, State College, Pa. Representing Engineering Experiment Station at the College.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, June, 1921.

sists of two buildings, the generating plant and the calorimeter building, as shown in Fig. 1. A description of the plant is given in Bulletin No. 30, Heat Transmission: Corkboard and Air Spaces, by A. J. Wood and E. F. Grundhofer of the Engineering Experiment Station, from which the following is taken.

The generating plant is a brick building 50 ft. by 18 ft. Its equipment, shown in Fig. 2, consists of a complete 3-ton experimental refrigerating system of the compression type, a variable speed circulating pump for carrying the brine to the calorimeter building and a motor-driven air compressor. The installation is such that the plant can be driven either by electricity or by steam since it includes motors, steam engines and a

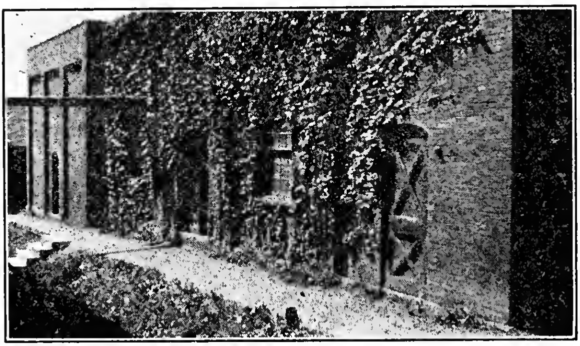

FIG. 1. THERMAL TESTING PLANT BUILDING, PENNSYLVANIA STATE COLLEGE, STATE COLLEGE, PA.

steam boiler. The observers' bench, Fig. 3, is also located in this building and contains the necessary voltmeters, ammeters, rheostats, etc., for measuring the electrical heat supplied to the plate and switches, a Wheatstone bridge and potentiometer for measuring the temperatures registered by the resistance thermometers and thermocouples in the calorimeter building.

The calorimeter building is a brick building 32 ft. square and contains, centrally located, an outer calorimeter room of corkboard construction, 20 ft. by 20 ft. by 10 ft. high. This room is fitted with 550 ft. of 1¼ in. brine coils distributed over three sides of the room. The brine flow can be regulated by means of a circulating pump in the generating plant, to hold the temperature within a variation of a few degrees. The test box formerly used in this room is a 5 ft. cube of 3 in. corkboard, with one side removable. Within this room a second calorimeter room, 10 ft. by 7 ft. by 10 ft. high, shown in Fig. 4, has been built and since its completion has been used for all heat transmission tests, including those on the new plate. It is of somewhat similar construction and is fitted with 690 ft. of 1¼ in. brine coils located on the four side walls and the ceiling. This inner room is so located that space is afforded in the large room for experimenting with large test specimens. This small calorimeter room is fitted with a Tagliabue thermostat which so regulates the flow of the brine that the room temperature can be held within 0.5 deg. fahr. and in several cases no difficulty has been experienced in maintaining a temperature which has varied 0.1 deg. fahr. for as long as 3 hours. The thermostat can be adjusted from the outside of this room to regulate at any desired temperature. The compressed air necessary for the operation of this instrument is supplied by the compressor in the generating plant.

Engineers are familiar with the heating plate for determining thermal conductivity as used by the Bureau of Standards. The distinctive difference in the use of the Bureau of Standards conductivity apparatus and the one here described, lies in the method of cooling the surface of the test

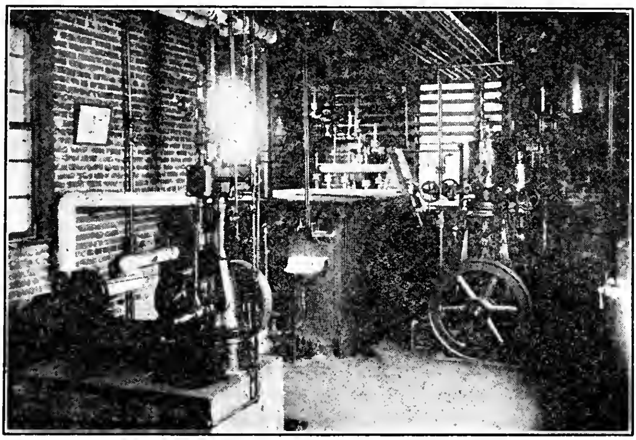

FIG. 2. REFRIGERATING PLANT AND BRINE CIRCULATING PUMP THE HEATING PLATE

blanks. In our work, one side of the blank is exposed to the air in a cold room which is kept at a uniform temperature. The plate method as thus adapted for our work can be used only when the room temperature can be controlled. By means of this set-up, both conductivity and surface transmission constants may be obtained at the same time.

Fig. 5 is a diagrammatic sketch of the assembled plate, showing the relative location of the blanks of the material to be tested. The main heating element, Fig. 6, is surrounded by an edge heating element, C, which supplies the heat loss from the four edges of the plate. The sections, 2 ft. square designated as A and A′, are the parts through which the heat losses are measured. The 5½ in. copper guard rings, B and B′ are placed around A and A′ and separated from them by ⅛ in. as shown in Fig. 7. The copper plates are insulated from the heating element by thin sheets of micanite.

Fig. 6 illustrates the winding of the two sections of the heating element. The main element is wound with 210.62 ft. of Advance ribbon, 138.57 ft. of which furnished heat to the two copper squares A and A′,

Fig. 5. The ribbon is ¼ in. wide and has a resistance of approximately 70 ohms per 1000 ft. The total resistance of the heating element is 15 ohms and will take a current of 7.3 amperes when connected directly to a 110 volt line. The maximum heat input per square foot of specimen is therefore 252 B.t.u. per hr. and is sufficient to give a temperature drop of approximately 30 deg. through concrete blanks 1 in. in thickness.

Temperatures of the plate and surfaces of the corkboard are determined by means of copper constantan thermocouples and a Leeds & Northrup potentiometer which can be read to 0.001 millivolt, corresponding to about 0.05 deg. The thermocouples were made of No. 30 silk-

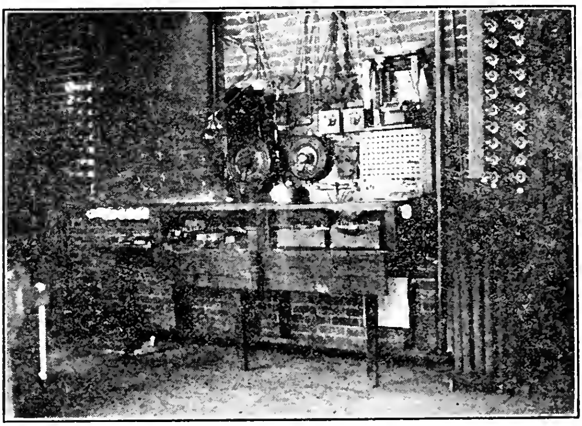

FIG. 3. TEMPERATURE MEASURING AND CONTROL INSTRUMENTS

covered wires and were calibrated at the Physical Laboratory of the United States Bureau of Mines to give an accuracy of ± 0.1 deg. fahr. Electrical leakage probably reduces somewhat this accuracy; however, ± 0.2 deg. can safely be assumed.

For determining the temperature of the copper plates, the constantan wires are soldered into the plates at points where the temperatures are to be taken, the copper of the plate itself being used as the positive element of the couple. Two of these couples are shown as I and J, Fig. 5, and E and F are the copper leads from the copper plates.

For maintaining the same temperature in the copper plates and the guard rings, constantan wires are connected across the spaces separating the two sets of plates. This gives a number of couples connected in parallel with their thermal junctions in the plates, the temperatures of which are to be kept the same. Any difference of temperature between the two center plates and the guard rings, results in a thermal E. M. F. between these two sets of plates. Temperatures of the hot and cold surfaces of the blanks are determined by placing the junctions of the couples under an extremely thin layer of the material. These couples are shown in Fig. 5 as K, L, M and N. The cold junctions of all the thermocouples are placed in contact with a Leeds & Northrup resistance thermometer in the cold room. Air temperatures are taken by means of Leeds & Northrup resistance thermometers suspended in the room about two feet from the surface of the blanks.

Leads from the thermocouples, resistance thermometers and heating elements are extended from the cold room to the temperature measuring and temperature control instruments which are placed on the observers bench, Fig. 3, located in an adjacent building.

Electric current is supplied from a motor generator set and the energy input measured by the "voltmeter-ammeter" method. Alternating cur-

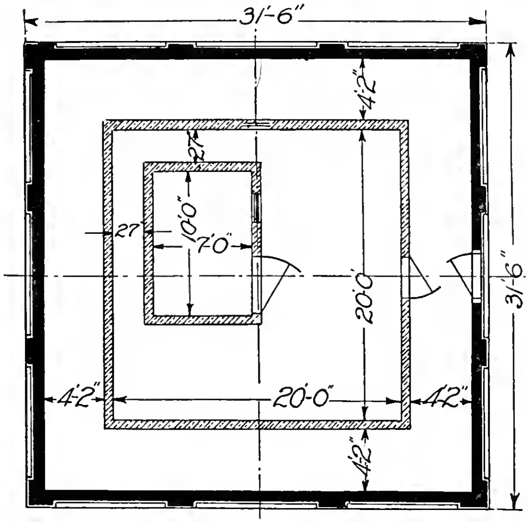

FIG. 4. PLAN OF CALORIMETER ROOM OF TESTING PLANT

rent is also available and used to maintain the temperature of the plate approximately constant during the night.

MANIPULATION

Preliminary to taking observations, the plate is heated up to the desired temperature by means of an electric current considerably greater than that calculated to give the desired conditions. The cold room is cooled by circulating the brine. As the temperature of the plate approaches the condition of the test, the current is reduced to the calculated value and maintained until a condition of equilibrium has been reached. The current in the auxiliary heating element is varied to hold the temperature of the center plates and the guard plates the same. Voltmeter and ammeter readings, temperatures of the hot and cold surfaces of the test material and the room and cold junction temperatures are recorded every fifteen minutes.

The time necessary to reach equilibrium condition varies with the temperature of the plate at the start. If a test is started in the morning with the plate cold it is found difficult to reach equilibrium conditions soon enough to obtain sufficient data in a day. To overcome this difficulty the plate is kept hot during the night by connecting it up with the alternating current.

In order to make a study of the temperature gradient over the surface of the plate and the hot and cold surfaces of the corkboard, 95 thermocouples were installed over these surfaces, in addition to those necessary to determine the constants and shown in Fig. 5. Fig. 9 shows the temperatures indicated by some of these couples during one of the tests, when a condition of equilibrium was maintained. The temperatures in ellipses were observed on the under side of the copper plate at the positions in-

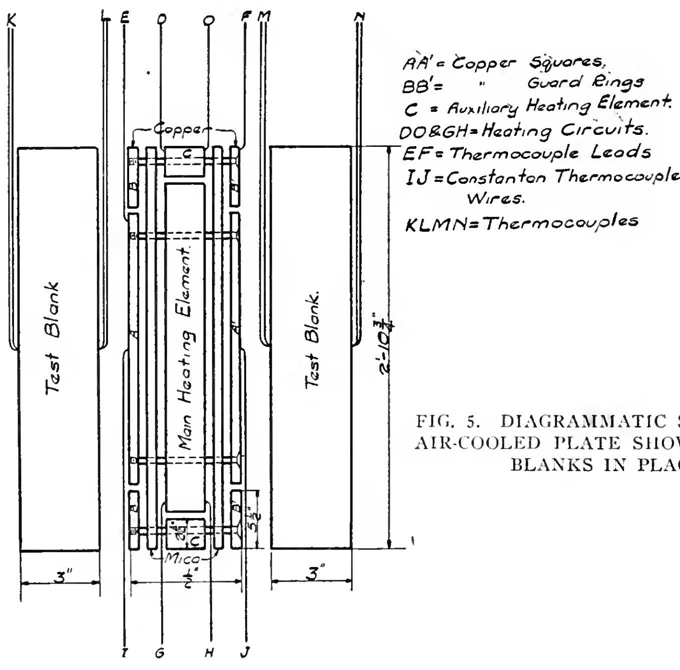

FIG. 5. DIAGRAMMATIC SKETCH OF AIR-COOLED PLATE SHOWING TEST BLANKS IN PLACE

dicated. The temperatures in rectangles were observed on the cold surface of the corkboard at the positions indicated.

The average temperature over the two foot center square is 101.9 deg. fahr., while that observed at the center is 0.2 deg. above the average and maximum variation from the average is 0.5 deg. The average temperature over the same area of the cold surface of the corkboard is 44.6 deg. fahr., which differs by 0.5 deg. from the center couple. The maximum variation from the average is 1.2 deg. fahr. Since these couples were placed under a thin shaving of the corkboard, which, though very thin, probably varied considerable in thickness, this variation might be expected. Further, the variation between the center couple and the average is not sufficient to influence the conductivity constant by an amount greater than its homogeneity. It is, however, enough to affect the surface transmission constant by a considerable amount, for the reason that the error is a much greater per cent of the total temperature difference between surface of the corkboard and the temperature of the air.

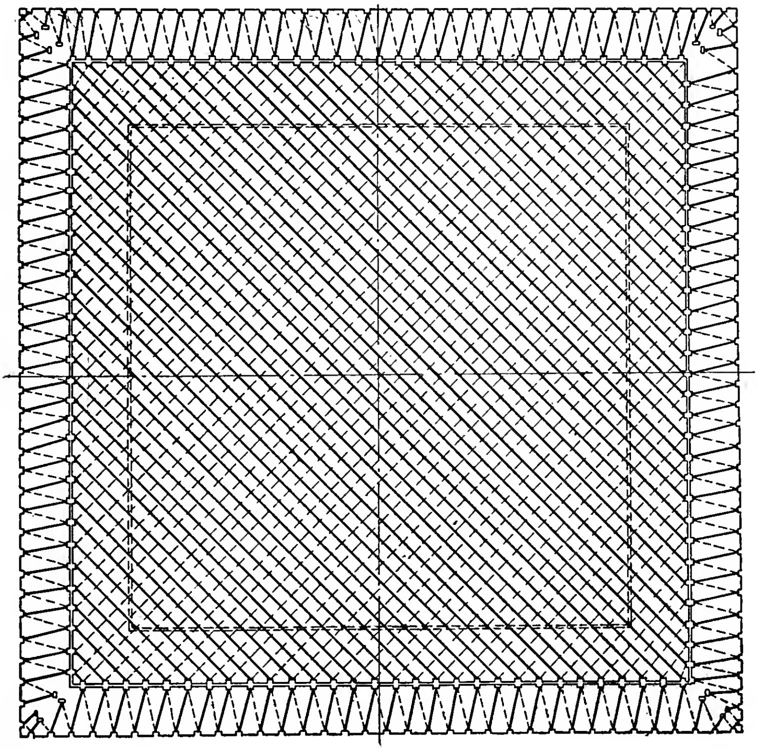

FIG. 6. MAIN HEATING ELEMENT AND EDGE HEATER, SHOWING METHOD OF APPLYING THE RESISTANCE RIBBON

METHODS OF CALCULATION

After a condition of equilibrium has been reached the heat passing through the specimen during any period of time is:

$$H = c\,A\,\frac{dt}{dS} \qquad (1)$$

where, H = heat passing through the area A of the specimen.

$\frac{dt}{dS}$ = temperature gradient through the specimen.

c = thermal conductivity of the material.

If t_1 is the temperature of the hot side of the specimen, t_2 the temperature of the cold side and S the thickness of the specimen, the temperature gradient through the material, $\frac{dt}{dS}$ may be expressed by, $\frac{t_1 - t_2}{S}$, and equation (1) becomes:

$$H = c\,A\,\frac{(t_1 - t_2)}{S} \tag{2}$$

c, the thermal conductivity of the material, then becomes:

$$c = \frac{H\,S}{A\,(t_1 - t_2)} \tag{3}$$

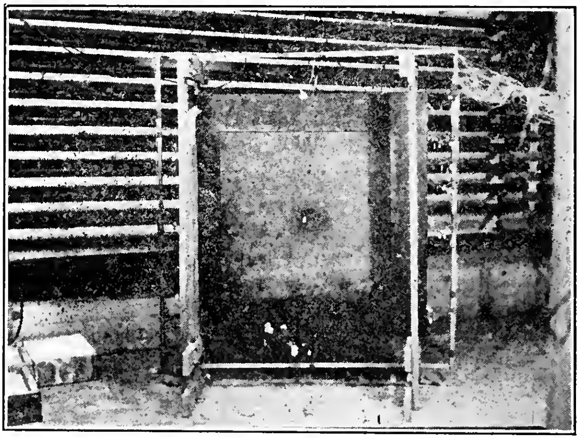

FIG. 7. HEATING PLATE AS COMPLETED READY FOR TEST BLANKS

In the case of our plate, H, the heat passing through the area A, of the specimen must be determined from the potential drop E across the heating element, the current I, through the heating element and the percentage of the heating element supplying heat to the area A. The heating element is wound with 210.62 ft. of Advance ribbon, 138.57 ft. of which furnishes heat to the 2 ft. squares. The heat passing through the specimen is then expressed by the formula:

$$H = \frac{138.57}{210.62}\,E\,I\,J \quad = 0.6579\,E\,I\,J \tag{4}$$

where J is the heat equivalent or 3.415 to express H in B.t.u. per hr. For our plate with corkboard A is 8 sq. ft. and S is 3 in.

Substituting these values in equation (3), we have:

$$c = \frac{.6579 \times 3 \times 3.415\,E\,I}{8\,(t_1 - t_2)} = .8425\,\frac{E\,I}{t_1 - t_2} \tag{5}$$

The surface transmission constant expressed in B.t.u. transmitted per sq. ft. per hr. per deg. difference of temperature between the surface,

and the air on the cold surface of the specimen, is given by k_2 in the equation: $H = k_2 A (t_2 - T_2)$ (6)
where T_2 is the temperature of the air.

$$k_2 = \frac{0.6579 \times 3.415 \, E \, I}{8 \, (t_2 - T_2)} = 0.2808 \frac{E \, I}{t_2 - T_2} \tag{7}$$

U, the total transmission for actual thickness expressed in B.t.u. per sq. ft. per hr. per deg. temperature difference between the air on the hot side and the air on the cold side of the blank, is given by the equation:

$$U = \frac{1}{\frac{S}{c} + \frac{1}{k_1} + \frac{1}{k_2}} \tag{8}$$

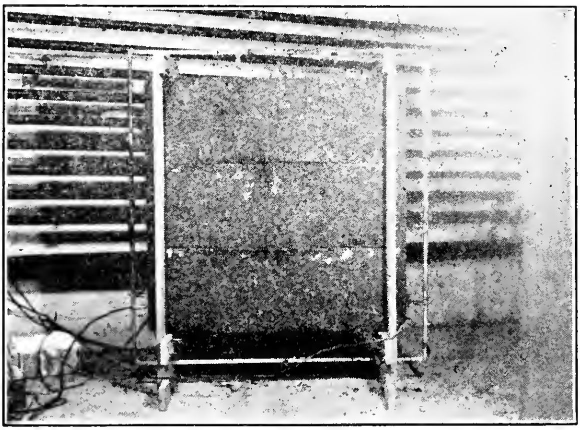

FIG. 8. AIR COOLED PLATE WITH CORKBOARD BLANKS IN POSITION FOR TEST

Where c is the conductivity expressed in B.t.u. per sq. ft. per hr. per in. thickness and k_1 is the surface transmission on the hot side of the material.

With the plate it is obviously impossible to determine k_1. As a matter of fact serious difficulties are met in determining this value by the box method. Although the inside and outside surface transmissions have been computed from temperature observations in a number of test boxes as used in different laboratories, the inside surface conditions have been so influenced by the method of testing peculiar to the individual type of box, that the constants for k_1 do not properly represent the conditions of the surface transmission. The errors have been caused by forced circulation, by radiation from the heating coils, by temperature observations taken too close to the surfaces or by other local conditions. While total transmission values, using k_1 and k_2, may have been expressed in some of these cases for purposes of comparisons, usually it has been considered more accurate to use the inside surface co-efficient as equal to the outside value, or $k_1 = k_2$. (See p. 41, *Bulletin 102, Eng. Exp. Station, University of Illinois* and p. 35, *Bulletin No. 30 Eng. Exp. Station, Penna. State College.*)

With the plate method adapted to obtain accurately values for outside surface transmission, as in the tests here recorded, the same data for computing total transmission is obtained as in the box methods as generally used up to the present time.

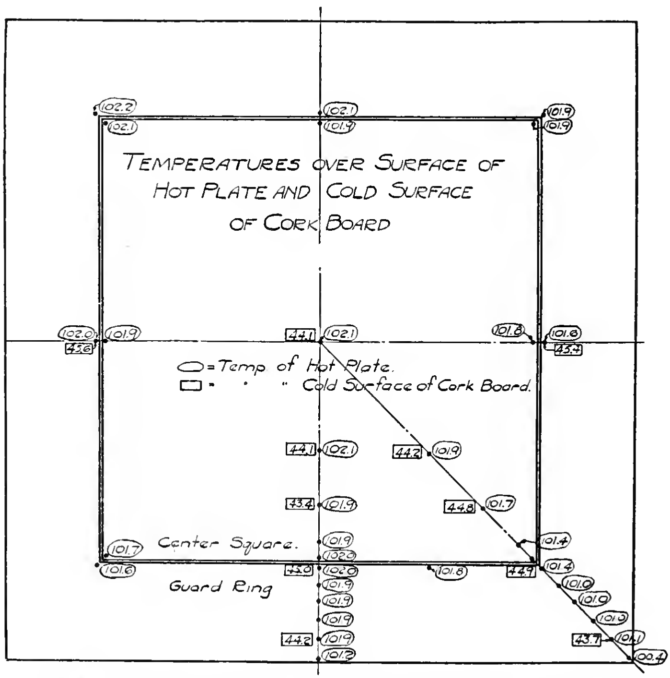

FIG. 9. TEMPERATURES OVER SURFACE OF HOT PLATE AND COLD SURFACE OF CORKBOARD

Making this assumption equation (8) becomes:

$$U = \frac{1}{\dfrac{S}{c} + \dfrac{2}{k_2}} = \frac{ck_2}{Sk_2 + 2c}$$

RESULTS

Table 1 contains the results of seven tests with temperature differences ranging from 52.4 to 71.5 deg. fahr. The values for c for the tests vary from 0.288 to 0.309 B.t.u. and have the average value of 0.301 B.t.u. The greatest variation from the average is 4.3 per cent for test 4-15-21, while the average variation from 0.301 is 1.7 per cent which is less than the homogeneity of the material.

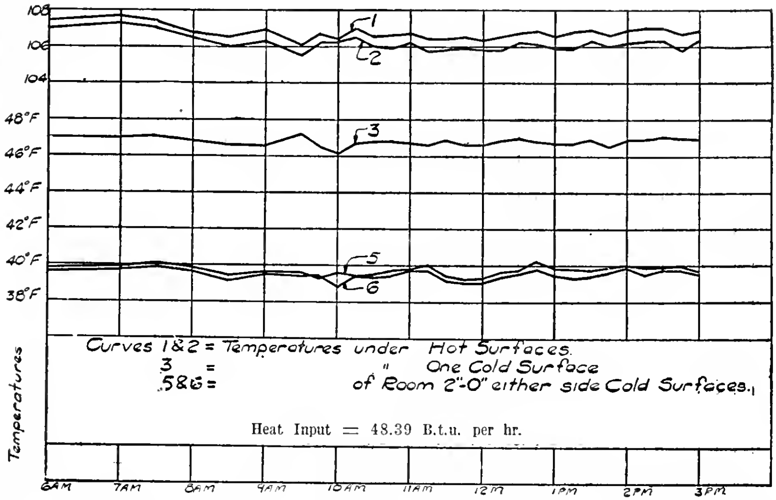

FIG. 10. REPRESENTATIVE TEST OF CORKBOARD WITH AIR COOLED PLATE

The average value for k_2 is 1.17 while the average variation from this value is 10 per cent. Although this error is rather large, it is not so much greater than one would expect from the magnitude of the values found for $t_2 - T_2$ and recorded in Table 1. The average of $t_2 - T_2$

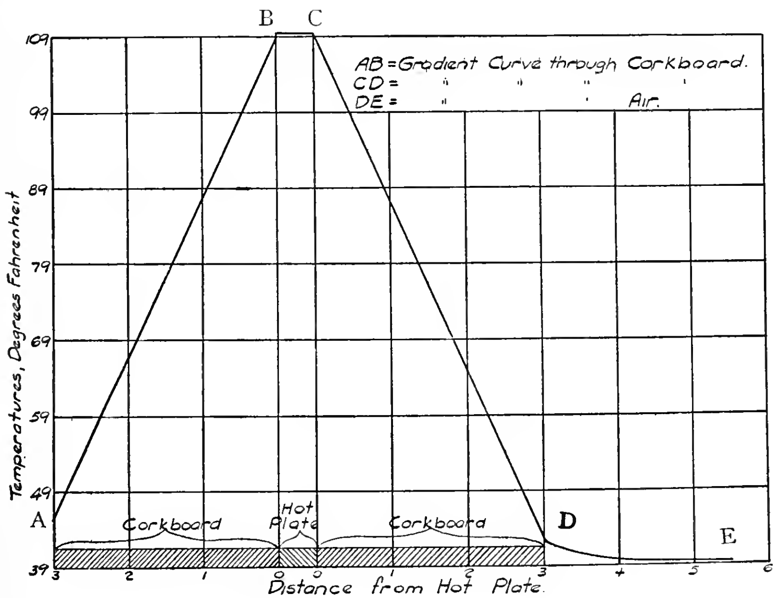

FIG. 11. TEMPERATURE GRADIENT CURVE THROUGH CORKBOARD BLANKS AND AIR

for these tests is 5.5 deg. fahr. An error of 0.2 deg. fahr. in determining each of the temperatures t_2 and T_2 would, if made in opposite directions, give an error of over 7 per cent in the value of k_2. However, this will not account for the 10 per cent variation in these values. Thermocouples are to be installed for the room temperatures and these should give more consistent values for k_2.

Test No. 4-7-21 which is representative of those recorded in Table 1, is plotted in Fig. 10. In Curves 1 and 2, the temperature of the two hot surfaces of the corkboard are plotted against time. Curve 3 is the temperature of one cold surface of the corkboard and Curves 5 and 6 are the two still air temperatures taken 2 ft. from the cold surfaces of cork-

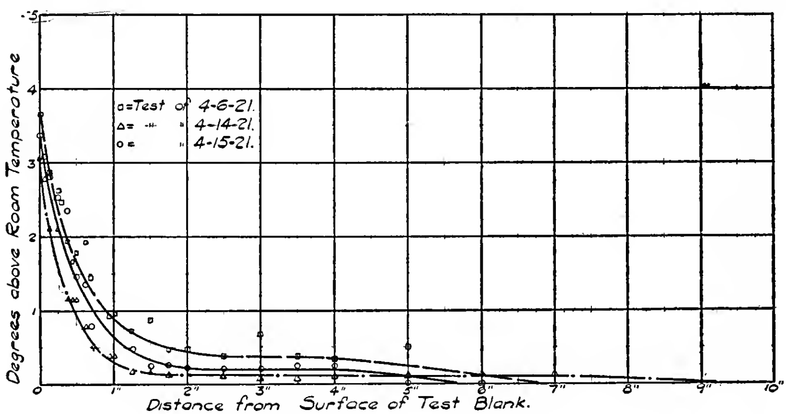

FIG. 12. TEMPERATURE GRADIENT CURVE THROUGH AIR

board, plotted against time. Conditions after 11 a. m. are very good for computing constants. The curve in Fig. 11 is the temperature gradient from the hot plate through the corkboard in both directions and out into the air on the right hand side. This curve is for test 4-14-21.

The temperatures along the gradient curve *D E* in air were determined by several of a series of 28 thermocouples extending a distance of 1 ft. away from the surface of the corkboard. The first eight of these thermocouples were placed 1/16 in. apart, the next four ⅛ in. apart, followed by four, ¼ in., four, ½ in. and eight, 1 in. apart.

Fig. 12 contains three temperature gradient curves through air, plotted on a larger scale than shown by *DE,* Fig. 11.

COMPARISON OF RESULTS

The average results of the seven tests recorded are shown in Column 1, Table 2. The results of work done at the Engineering Experiment Station, State College, using their small cubical box and published in Engineering Experiment *Bulletin No. 30,* are shown in the second column. Results from tests conducted at the Engineering Experiment Station, University of Illinois and published in *Bulletin No. 102* are given in Column 3. The last two columns contain results from tests conducted at the United States Bureau of Standards, using a water cooled plate.

The values in the last two columns are for corkboard of different densities. The value of c for our tests lies between the two values given by the Bureau of Standards and in about the proper position considering the density. The values given by the Pennsylvania State College and University of Illinois Experiment Stations are each higher. The values obtained for surface transmission and total transmission from these tests are also lower than those obtained by either the Pennsylvania State College or the University of Illinois.

ACKNOWLEDGMENTS

The authors appreciate the cooperation of the Bureau of Standards in connection with the designs and would also acknowledge the effective work of Prof. E. F. Grundhofer of the Engineering Experiment Station during the construction and testing of the plate.

SYMBOLS USED IN TABLE 1

H = Total heat input B.t.u. per hr. for measured area.
t_1 = Temperature of hot surface of corkboard.
t_2 = Temperature of cold surface of corkboard.
T_2 = Temperature of still air two ft. from cold surface of corkboard.
k_1 = Surface transmission constant hot side of corkboard B.t.u. per hr. per sq. ft. per deg. fahr.
k_2 = Surface transmission constant cold side of corkboard B.t.u. per hr. per sq. ft. per deg. fahr.
C = Conduction B.t.u. per hr. per sq. ft. per deg. fahr. per actual thickness.
c = Conductivity B.t.u. per hr. per sq. ft. per deg. fahr. per in. thickness.
U = Total transmission B.t.u. per hr. per sq. ft. per deg. fahr. per actual thickness.

TABLE 1. RESULTS OF TESTS ON 3 IN. CORKBOARD AT APPROXIMATELY 60 DEG. FAHR. DIFFERENCE

	H	t_1	t_2	t_1—t_2	T_2	t_2-T_2	k_2	C	c	c (24 hr.)
4- 1-21	40.98	97.3	44.9	52.4	39.8	5.1	1.19	.099	.297	7.13
4- 7-21	48.40	106.8	46.7	60.1	39.6			.101	.303	7.27
4- 8-21	51.20	109.5	46.5	63.0	40.0	6.5	.97	.102	.306	7.40
4-11-21	51.68	108.6	44.5	64.1	40.0	4.5	1.47	.101	.303	7.27
4-14-21	51.68	109.6	44.5	65.1	39.5	5.0	1.28	.099	.297	7.13
4-15-21	53.40	107.1	45.6	71.5	39.2	6.4	1.08	.096	.288	6.91
4-19-21	48.93	97.1	37.7	59.4	32.2	5.5	1.12	.103	.309	7.42
Average Results							1.18	.100	.301	7.22

TABLE 2. COMPARISON OF RESULTS FROM AIR COOLED PLATE WITH OTHER RESULTS

	1 Air Cooled Plate	2 Cubical Box P. S. C. Bulletin No. 30	3 U. of Ill. Bulletin No. 102	4 Water Cooled Plate U. S. B. S.	5 Water Cooled Plate U. S. B. S.
C	.100	.109	.106	.101	.090
c	.301	.327	.32	.304	.271
c (24 hr.)	7.22	7.85	7.68	7.3	6.5
k_2	1.18	1.20	1.25		
k_2 (24 hr.)	28.1	28.8	30.0		
U	.086	.097	.090		
U (24 hr.)	2.05	2.33	2.09		
Density of corkboard, lb. per cu. ft.	9.3	9.2		9.9	6.9

NOTE: First series of results using new air cooled plate. Density of corkboard tested, 9.3 lb. per cu. ft.

No. 612

COMPARATIVE TESTS OF AIR DUSTINESS WITH THE DUST COUNTER, KONIMETER AND SUGAR TUBE

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Non-Members

INTRODUCTION

Studies of air dustiness in mineral industries as related to the health of miners and others have been conducted by the Bureau of Mines for many years. More recently a comparative study of methods of determining rock dust in air was undertaken with the object of securing better interpretation of analytical results and data upon which results obtained by one method might be converted into those by another. A paper entitled: Efficiency of the Palmer Apparatus for Determining Dust in Air, that describes the method of dust determination by washing air with water, has previously been published.[3] A Study of the Sugar Tube Method of Determining Rock Dust in Air, describing the method of determining dust in air by filtering through sugar, is in process of publication by the Bureau of Mines; a short digest of this has been presented.[4] The present paper is a preliminary statement of some results obtained in the investigation of dustiness of the air in granite working plants at Barre, Vt., and for the present the figures are to be regarded as tentative. In this work comparative tests of dustiness were made with the sugar tube, the Hill dust counter,[5] and the "circular" konimeter.[6] The latter two instruments use the method of impinging a measured volume of air at high velocity against a sticky plate. The particles of dust caught are later counted under a microscope with a magnification of 50 diameters for the dust counter and 200 diameters for the konimeter; these magnifications are convenient for fitting the two sizes of dust spots to the field of microscopic vision. Dust caught by the sugar tube is counted at 110 diameters magnification, which has been the standard for the sugar

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[3] *Journal*, American Society of Heating and Ventilating Engineers, Nov., 1920, pp. 687-700.

[4] *Journal*, American Society of Heating and Ventilating Engineers, Mar. 1921, pp. 119-123.

[5] Quantitative Determination of Air Dust, by Dr. E. V. Hill. *Journal*, American Society of Heating and Ventilating Engineers, June, 1917, pp. 23-33.

[6] Final report of the Miner's Phthisis Prevention Committee: *Union of South Africa*, Johannesburg, Jan. 10, 1919, p. 110.

Published with permission of the Director, U. S. Bureau of Mines. Experiments carried out at Industrial Gas Laboratory, Pittsburgh Experimental Station, U. S. Bureau of Mines, Pittsburgh, Pa.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, O., June, 1921.

tube method of determination. Results of counts are expressed in numbers of particles per cu. in. or millions of particles per cu. ft.

Averages of all the tests at Barre show that the konimeter method indicates about 13 times as many dust particles as the dust counter, and the sugar tube about 7 times as many as the konimeter; the factor for the

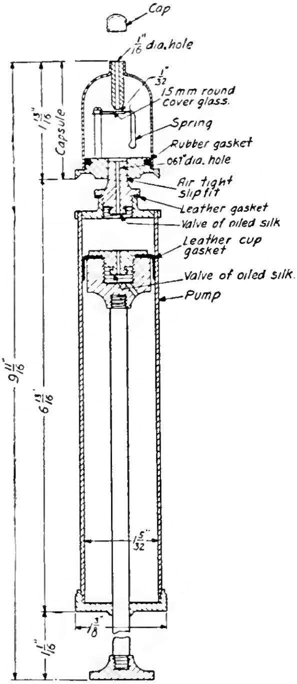

FIG. 1. DETAILS OF THE DUST COUNTER

dust counter and sugar tube is about 83. Thus, very large differences occur among the results by the different methods.

DESCRIPTION OF THE INSTRUMENTS

Dust Counter. The dust counter shown in diagramatic form in Fig. 1 consists of a small cylinder and piston arranged for drawing a volume of air rapidly through a nozzle onto a sticky plate. Fig. 2, pictures it together with the hand case for carrying the instrument and accessories. To prepare the dust counter for use a microscope cover glass is coated with a thin film of Canada balsam and placed in the end piece or capsule where it is supported on a small stand at a distance of 1/32 (.03125) in. beneath the nozzle. Three capsules with the instrument may be prepared in advance for the sampling and carried in the case. The capsules are readily attached to the pump as needed by an air tight slip fit. One moderately rapid pull of the piston is sufficient for taking a sample in a dusty atmosphere; two, three or as many as ten pulls are given per sample in less dusty air. Calibration of the volume per stroke gave 5.28 cu. in. (86.5 cu. cm.).

The cover glasses with dust samples are permanently mounted upon a microscope slide with a cloth washer or ring between to prevent contact of the dust spot with the slide. Counting is done under a cross-ruled screen for reference lines.

Fifty diameters magnification is used since this is the highest power that conveniently allows the observer to bring the entire spot, approximately 1/16 in. in diameter, into the field of vision at one time. The

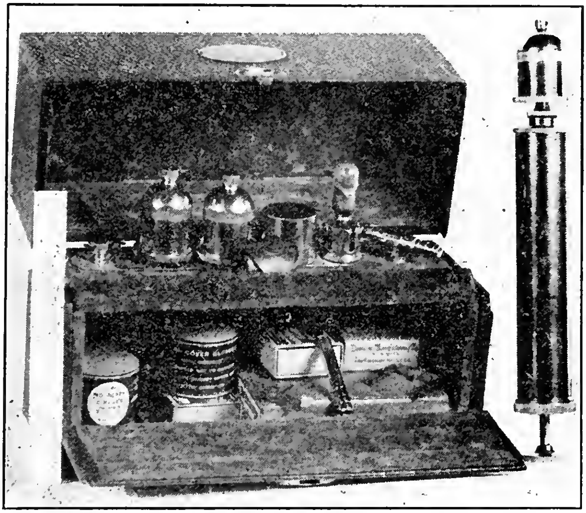

FIG. 2. DUST COUNTER WITH CARRYING CASE AND ACCESSORIES

result of the count multiplied by a volume factor for the instrument gives the number of dust particles per cu. cm. or cu. ft.

The dust counter is especially useful where permanent samples of the air dust are desired. It is also well adapted for dusts that are soluble in water or that tend to conglomerate or disintegrate in water solutions.

Konimeter. The *circular* konimeter used in this work was imported from South Africa where it was developed, and was of most recent design. Fig. 3 gives the essential details. Fig. 4 shows it in use. In principle the konimeter is much like the dust counter, but instead of actuating the piston by hand it is propelled by a spring released by a trigger. The air passes through a nozzle .0225 in. in diameter, and discharges onto a circular glass plate covered with a thin coat of vaseline placed .02 in. (.5 mm) away. It then passes to a tube leading to the pump cylinder. The capacity of the pump is .659 cu. in. or 10.8 cu. cm. The glass plate is cemented at its edge to a toothed brass ring that engages a worm pinion. The pinion shaft extends outside the instrument and on it is a small cross lever; turning this brings another point on the glass plate opposite the nozzle. The ring is numbered in 29 sectors so that this number of samples may be taken on a single plate.

To count the particles in dust spots the glass plate is removed from the instrument and placed under a microscope at 200 magnifications. A grating facilitates the counting. In this case 200 diameters is employed because the small diameter of the spot, about 1/40 in., allows it to come within the field of vision at the higher magnification. The results of the count are multiplied by a factor to give the number of particles of

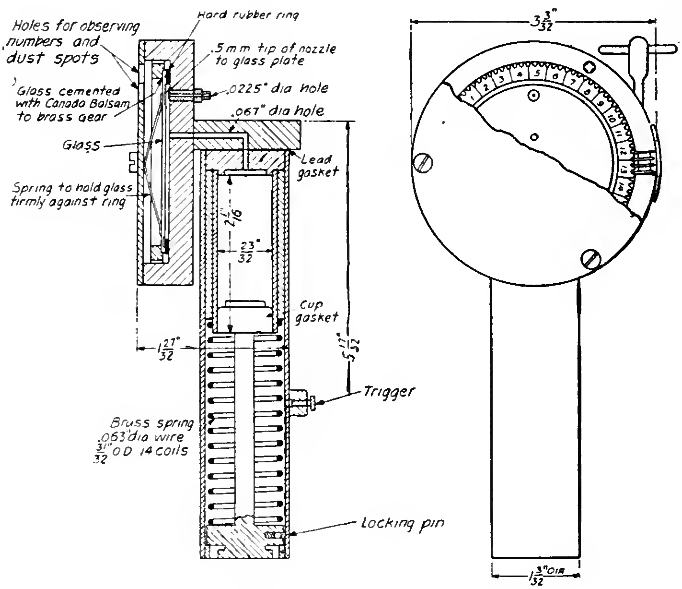

FIG. 3. DETAILS OF THE KONIMETER

dust in unit volume. When the glass plate is cleaned and again coated with vaseline it is ready for the next set of dust samples. The advantage of the konimeter is the large number of samples that may be very conveniently and speedily taken on one plate.

Sugar Tube. The sugar tube is shown in use in Fig. 5. It consists of a glass cylinder 2⅜ in. in diameter containing a screen of monel wire that supports a layer of granulated sugar 1⅝ in. deep and weighing 100 gm. A noncollapsible rubber tube connects the sugar tube to a calibrated foot pump with which about 15 cu. ft. of air are drawn through the sugar in a 15 min. sampling period. A motor-driven pump with a resistance flowmeter for measuring air has been designed for eliminating the laborious pumping where electric power is available.

The sugar with its dust is sent to a laboratory where the sugar is dissolved in pure water and diluted to a definite volume. An aliquot portion of this is placed in a Sedgwick Rafter cell for counting the insoluble dust particles. One cu. mm. of solution is observed after the

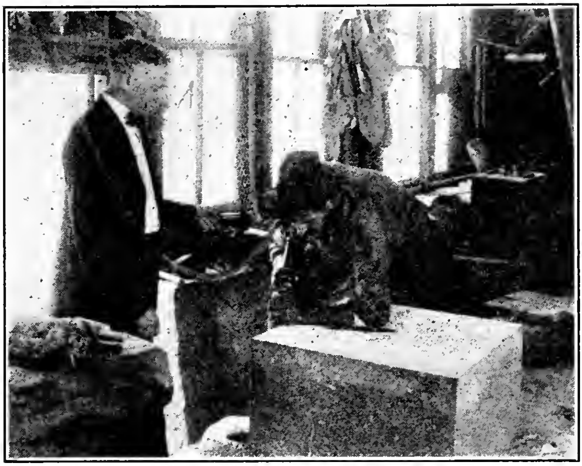

FIG. 4. TAKING A DUST SAMPLE WITH THE KONIMETER

dust has settled under a cross line grating at 110 diameters magnification. From the number of particles counted, and the volume of air sample, the number of particles in unit volume of air is figured. The remaining sugar solution may be filtered through a filter of pure paper, the paper

FIG. 5. SUGAR TUBE IN USE FOR SAMPLING DUST IN AIR

burned off and the non-combustible dust weighed. Results of weight determinations are expressed in milligrams of dust, per cubic meter cu. ft. of air. The rock dusts used for this series of tests are h well by the sugar tube method; this method is not applicable to dus hat dissolve in water or those that will disintegrate into smaller particles or combine into larger ones.

COMPARISON OF RESULTS WITH THE THREE INSTRUMENTS

Results obtained at two of the five plants are given graphically in Figs. 6 and 7. The total numbers of dust particles of all sizes are represented as ordinates and the serial number of the samples abscissæ. Fifty

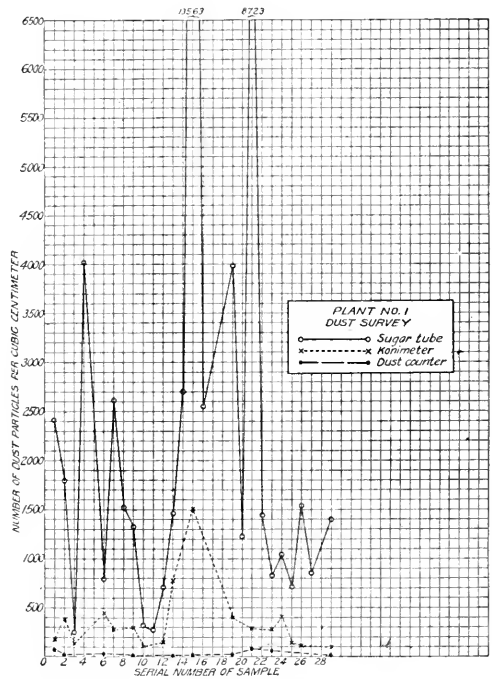

FIG. 6. CHART SHOWING VARIATION IN DUSTINESS OF AIR IN GRANITE PLANT NO. 1

microns was approximately the largest diameter of particle, while the smallest particles counted, varied from ½ to 2 microns according to the magnification. (1 micron equals 1/25000 in.) A large majority of particles were under 10 microns in diameter and the numbers increased as the size diminished.

The dust counter and konimeter give results of instantaneous grab samples whereas the sugar tube represents an average dustiness over a 15 min. period. In all cases the sugar tube samples were taken first, and the other two samples were taken at the end of the period. The sampler always endeavored to have corresponding samples by the different instruments representative of the same conditions. Samples were taken in various parts of plants with the object of making a survey including all conditions of dustiness to which workmen may be exposed.

Very large variations in degree of dustiness at any one plant are apparent. In all but two situations the numbers of particles indicated by the sugar were much greater than those by the konimeter, and the numbers indicated by the konimeter were greater than those by the dust counter. The lower result with the dust counter is due partly to inability to observe some of the smaller particles at the magnification of 50 diame-

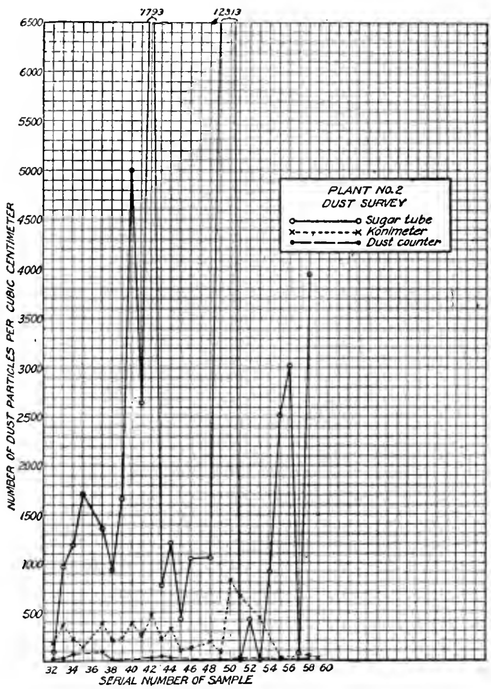

FIG. 7. CHART SHOWING VARIATION IN DUSTINESS OF THE AIR IN GRANITE PLANT NO. 2

ters. On the other hand, the sugar tube particles counted at 110 diameters were much more than those with the konimeter counted at 200 diameters.

Relations between the numbers of particles as determined by the different instruments expressed as ratios or factors are quite variable among single sets of tests. This is not surprising considering the fact that non-uniformity of dustiness is apparent by gross observation with the naked eye. Evidently a large number of observations must be considered statistically for drawing conclusions. Hence, the ratios of summations of particles from comparative tests at each plant have been taken. The plant averages (see Fig. 8), show fair agreement. Averaging the ratios from all tests gives these factors: To convert dustiness as determined by dust counter to that by sugar tube, multiply by 83; konimeter to sugar tube, 6.7; dust counter to konimeter, 13.

It is obvious from the preceding figures that a number of samples should be taken and results averaged to obtain reliable information regarding dustiness; and the reliance that may be placed, increases with the number of terms averaged.

SUMMARY AND CONCLUSIONS

Comparative tests of air dustiness with the dust counter, konimeter and sugar tube were made in granite working plants. Results, which for the present are tentative, show that the sugar tube indicates 83 times as

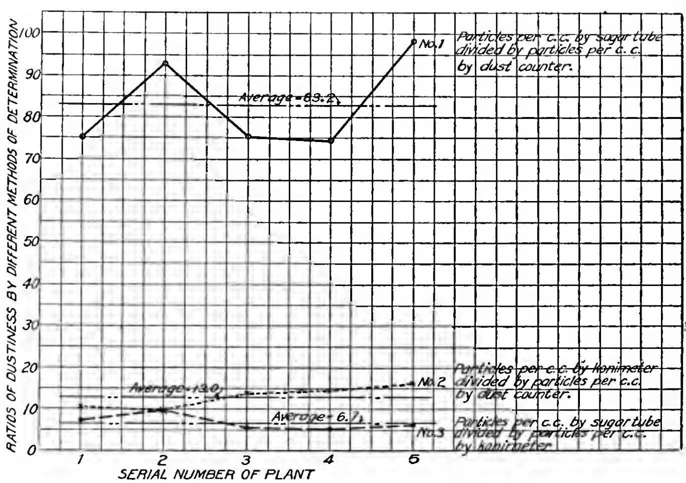

FIG. 8. CHART SHOWING RATIOS OF DUST PARTICLES PER UNIT VOLUME

much dust as the dust counter, and 6.7 times as much as the konimeter; the konimeter indicates 13 times as much as the dust counter. In determining dustiness, a number of samples should be taken and the results averaged.

ACKNOWLEDGMENTS

The authors are indebted to O. W. Armspach of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for many valuable suggestions applied in the conduct of this work, and to A. C. Fieldner, Supervising Chemist of the Pittsburgh Experiment Station of the Bureau of Mines under whose general direction this work was done.

DISCUSSION

O. W. ARMSPACH: There is one point I would like to bring out, and that is, that a dust determination with any instrument depends upon two things, two absolutely independent things. The *first* is that the final count will depend upon the *efficiency of the instrument to retain the dust.* The *second* is that the final count will depend upon the *accuracy of the method of counting* and this is independent of the efficiency of the machine. There are two different divisions there. One is the efficiency of the instrument and the other is the accuracy of counting the dust retained by the instrument. Both of these enter materially into the final figure arrived at, and the number of particles reported is not necessarily a measure of the efficiency of the dust counting device.

I am interested to see that the figures presented with respect to the dust count determined with the sugar tube agree very closely with the figures reported before this Society at the June meeting, 1919, and published on page 289 in the 1919 TRANSACTIONS. There we gave a figure of from one to two hundred for a multiplication factor to be applied to the sugar-tube method using a counting cell. The counting cell will give a much higher count than that obtained with other instruments. Those interested in the subject can look up that report and see what our explanation of the reason for this is.

Mr. Katz's paper is very valuable in that it is a start on getting comparative figures so that dust counts in different places as taken by different instruments, can be compared. That is a thing that we must come to because it is of no value whatever to compare dust conditions in different industries until they are placed on the same basis.

L. A. SCIPIO: I would like to add a word here. It would seem that there has been enough work done to show the difference between the accuracy of the sugar tube and the other instruments commonly used and even this is limited to testing only inorganic dust. Any instrument that depends upon a count being made by means of a microscope is unsatisfactory because of the human element entering into it. If it were possible to devise an instrument through which a large quantity of air could be passed and the dust collected and weighed it would tend to eliminate most of the difficulties now attending dust determinations.

AUTHOR: In regard to weight, the particles with which we are concerned in investigating dust in mines range in size from about one-half a micron in diameter—a micron is 1/25,000th part of an in.—up to 50 microns in diameter. For a range from ½ micron to 50 microns in diameter a particle can vary in weight as 1 compared to 1,000,000 so that the inclusion of one of the larger particles in our collection will amount to as much as a million or more of the smaller particles. For this reason, we have always felt that it was necessary to have both count and weight in order to arrive at a safe conclusion.

In Memoriam

Names	*Joined the Society*	*Died*
John L. Moyer	June, 1914	January 20, 1921
W. F. Hughes	May, 1920	February 12, 1921
Wm. G. McPherson	December, 1901	February 17, 1921
Robert Ross Jones	October, 1915	March 19, 1921
John Boylston, Jr.	September, 1920	April 27, 1921
James A. Bendure	December, 1913	May 31, 1921
Frank A. Dow	August, 1918	June 2, 1921
Nathaniel W. Hunt	March, 1920	June 25, 1921
Charles Morrison	June, 1910	August 3, 1921
Clate B. Vinton	July, 1920	September 22, 1921
William K. Downey	July, 1901	October 1, 1921
J. Prator Dugger	December, 1906	October 29, 1921
Stephen F. McDonald	June, 1919	October 31, 1921
Joseph H. Bowers	June, 1919	November 4, 1921
Edward S. Berry	June, 1904	December 9, 1921

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